Chapter 7
Energy Conversion

7.1 Energy Conversion

In most processes, energy is constantly changing from one form to another. This is called *energy conversion*. Examples include the living systems converting the solar energy to chemical energy by synthesizing food from water and carbon dioxide through the photosynthesis. The mechanical energy of a waterfall can also be converted to electromagnetic energy in a generator. An internal combustion engine converts the potential chemical energy in gasoline into heat, which is then transformed into the kinetic energy that moves a vehicle. A solar cell converts solar radiation into electrical energy that can then be used to light a bulb or power a computer. The energy that enters a conversion device or a process is turned into other forms of energy, so an equal quantity of energy before and after is maintained. That means the energy is conserved during any form of energy conversion in a system.

Energy is most usable where it is most concentrated as in highly structured chemical bonds in gasoline and sugar. All other forms of energy may be completely converted to heat, but the conversion of heat to other forms of energy cannot be complete. Due to inefficiencies such as friction, heat loss, and other factors, thermal efficiencies of energy conversion are typically much less than 100%. For example, only 35–40% of the heat can be converted to electricity in a steam power plant and a typical gasoline automobile engine operates at around 25% efficiency. With each energy conversion, a part of the energy is lost usually in less useful and dispersed form of thermal energy, as illustrated in Fig. 7.1. For example, a light bulb converts only around 10% of electrical energy to light and the remaining is converted to heat, which is difficult to use to do work. This shows that there are limitations to the efficiency for energy conversion. Only a part of energy may be converted to useful work and the remainder of the energy must be reserved to be transferred to a thermal reservoir at a lower temperature [2, 4, 5, 13].

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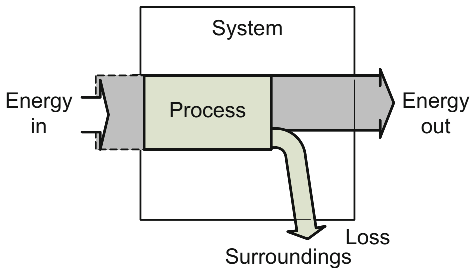


Fig. 7.1 Schematic of the energy usage and conversion in a process. Output energy at a new form is always lower than input energy. The total energy input is recovered in various other forms and hence the energy is conserved

There are many different processes and devices that convert energy from one form to another. Table 7.1 shows a short list of such processes and devices. When electric current flows in a circuit, it can transfer energy to do work. Devices convert electrical energy into many useful forms, such as heat (electric heaters), light (light bulbs), motion (electric motors), sound (loudspeaker), and information technological processes (computers). Electric energy is one of the most useful forms of output energy, which can be produced by various mechanical and/or chemical devices. There are seven fundamental methods of directly transforming other forms of energy into electrical energy [5, 31]:

- *Static electricity* is produced from the physical separation and transport of charge. Electrons are mechanically separated and transported to increase their electric potential and imbalance of positive and negative charges leads to static electricity. For example, lightning is a natural example of static discharge. Also low conductivity fluids in pipes can build up static electricity.
- *Electromagnetic induction*, where an electrical generator, dynamo, or alternator transforms kinetic energy into electricity. Almost all commercial electrical generation is done using electromagnetic induction, in which mechanical energy forces an electrical generator to rotate. There are many different methods of developing the mechanical energy, including heat engines, hydro, wind, and tidal power.
- *Electrochemistry* is the direct transformation of chemical energy into electricity, as in a battery, fuel cell, or nerve impulse.
- *Photoelectric effect* is the transformation of light into electrical energy, as in solar cells.
- *Thermoelectric effect* is the direct conversion of temperature differences to electricity, as in thermocouples, thermopiles, and thermionic converters.
- *Piezoelectric effect* is the electricity from the mechanical strain of electrically anisotropic molecules or crystals.
- *Nuclear transformation* is the creation and acceleration of charged particles. The direct conversion of nuclear energy to electricity by beta decay is used only on a small scale.

Table 7.1 Some of the processes and devices converting energy from one form to another

Process and device	Energy in	Useful energy output
Steam engine	Heat	Mechanical energy
Photosynthesis	Solar energy	Chemical energy
Hydroelectric dams	Gravitational potential energy	Electric energy
Windmills	Mechanical energy	Electric energy
Electric generator	Mechanical energy	Electric energy
Diesel or petrol engine	Chemical energy	Mechanical energy
Electric motor	Electric energy	Mechanical energy
Fuel cells	Chemical energy	Electric energy
Battery	Chemical energy	Electric energy
Electric bulb	Electric energy	Heat & light
Resistance heater	Electric energy	Heat
Ocean thermal power	Heat	Electric energy
Bioluminescence	Chemical energy	Light energy
Nerve impulse	Chemical energy	Electrical energy
Muscular activity	Chemical energy	Mechanical energy
Geothermal power	Heat	Electric energy
Wave power	Mechanical energy	Electric energy
Friction	Kinetic energy	Heat
Thermoelectric	Heat	Electric energy
Piezoelectrics	Strain	Electric energy

Granet and Bluestein [13], Chih [4]

7.2 Series of Energy Conversions

It takes a whole series of energy conversions in various processes before a useful form of energy becomes available. For example, when you use your computer, there are several following energy conversion processes involved:

- Chemical energy stored in coal is released as heat when the coal is burned.
- The heat is used to produce steam which is converted into mechanical energy in a turbine.
- The generator converts mechanical energy into electric energy that travels through the power lines into your home.
- From the power outlet at home, the computer receives that electric energy.

In a conventional internal combustion engine, these energy transformations are involved:

- Potential energy in the fuel is converted to kinetic energy of expanding gas after combustion.
- Kinetic energy of expanding gas is converted to piston movement and hence to rotary crankshaft movement.
- Rotary crankshaft movement is passed into the transmission assembly to drive the wheels of a car.

7.3 Conversion of Chemical Energy of Fuel to Heat

Chemical energy of a fuel is converted to heat during a combustion or oxidation reaction. Combustion reactions are exothermic and release heat. Heat of combustion is the same as heat of reaction for a combustion reaction, which is discussed in Sect. 4.3.4. For example, standard heat of combustion of propane (C_3H_8) can be estimated by using the standard heats of formation tabulated in Table C1. The combustion reaction of 1 mol propane is:

$$C_3H_8(g) + 5O_2(g) \rightarrow 3CO_2(g) + 4H_2O(g) + \text{Heat} \tag{7.1}$$

When the products and reactants are at their standard states (ideal-gas state at 1 bar and 25°C) the standard heat of reaction is expressed by

$$\Delta H_r^o = \sum_i v_i \Delta H_{fi}^o = v_{CO_2} \Delta H_{fCO_2}^o + v_{H_2O} \Delta H_{fH_2O}^o + v_{C_3H_8} \Delta H_{fC_3H_8}^o \tag{7.2}$$

where v_i is the stoichiometric coefficient of substance i, which is positive for a product and negative for a reactant. For example, for the reaction above $v_{CO_2} = 3$, $v_{H_2O} = 4$, $v_{C_3H_8} = -1$.

7.3.1 Heating Value of a Fuel

The *heating value* of a fuel is the amount of heat released during combustion [35]. It is measured in units of energy per unit of the substance, usually mass, such as: kJ/kg, Btu/m^3, or kcal/kg. The heating values for fuels are expressed as the higher heating value (HHV), lower heating value (LHV), or gross heating value (GHV). HHV is determined by bringing all the products of combustion back to the original precombustion temperature, and in particular condensing any water vapor produced. The combustion process of a fuel consisting of carbon and hydrogen can be approximately represented by

$$[C + H](\text{Fuel}) + [O_2 + N_2](\text{Air}) \rightarrow CO_2 + H_2O(\text{liquid}) + N_2 + \text{Heat (HHV)} \tag{7.3}$$

$$[C + H](\text{Fuel}) + [O_2 + N_2](\text{Air}) \rightarrow CO_2 + H_2O(\text{vapor}) + N_2 + \text{Heat (LHV)} \tag{7.4}$$

where C = Carbon, H = Hydrogen, O = Oxygen, and N = Nitrogen. Gross heating value takes into account the heat used to vaporize the water during the combustion reaction as well as the water existing within the fuel before it has been burned. This value is especially important for fuels such as wood or coal, which contains some amount of water prior to burning. Heating value is also discussed in Sect. 2.4. Higher and lower heating values of some common fuels, and energy density of some fuels are tabulated in Tables 2.7, 2.8, and 2.9.

Lower heating value (or *net calorific value*) is determined by subtracting the heat of vaporization of the water vapor from the higher heating value. A common method of relating higher heating value to lower heating value is:

$$\begin{aligned}&\mathrm{LHV} = \mathrm{HHV} - \left(\Delta H_{\mathrm{vap}}\right)\left(n_{\mathrm{H_2O,\,out}}/n_{\mathrm{fuel,\,in}}\right)\left(MW_{\mathrm{H_2O,\,out}}/MW_{\mathrm{fuel,\,in}}\right)\\&\text{or}\\&\mathrm{LHV} = \mathrm{HHV} - \left(\Delta H_{\mathrm{vap}}\right)\left(m_{\mathrm{H_2O,\,out}}/m_{\mathrm{fuel,\,in}}\right)\end{aligned} \tag{7.5}$$

where ΔH_{vap} is the heat of vaporization of water (in kJ/kg or Btu/lb), $n_{\mathrm{H_2O,\,out}}$ is the moles of water vaporized and $n_{\mathrm{fuel,in}}$ is the number of moles of fuel combusted, $MW_{\mathrm{H_2O}}$ is the molecular weight of water, and MW_{fuel} is the molecular weight of fuel. Example 7.1 illustrates the estimation of lower heating value from higher heating value, while Example 7.2 shows the estimation of heating values from the standard heat of combustion.

Example 7.1 Estimation of lower heating value from higher heating value
Higher heating value of natural gas is measured as 23,875 Btu/lb around room temperature (70°F). Convert this higher heating value to lower heating value. Heat of vaporization of water: ΔH_{vap} = 1,055 Btu/lb (70 °F).

Solution:
Assume that the natural gas is represented by methane CH_4.

The combustion of methane: $CH_4 + 3/2O_2 \rightarrow CO_2 + 2H_2O$
Higher heating value: HHV(methane) = 23875 Btu/lb,
From Table A1:
$MW_{\mathrm{H_2O}}$ = (H_2) + $1/2(O_2)$ = 2 + 16 = 18 lb/lbmol
$MW_{\mathrm{CH_4}}$ = (C) + 4(H) = 12 + 4 = 16 lb/lbmol

$n_{\mathrm{CH_4}}$ = 1 lbmol and $n_{\mathrm{H_2O}}$ = 2 lbmol and $n_{\mathrm{H_2O}}/n_{\mathrm{CH_4}}$ = 2/1

Heat of vaporization: ΔH_{vap} = 1055 Btu/lb (70°F)

Lower heating value:
LHV = HHV − $(\Delta H_{\mathrm{vap}})$ $(n_{\mathrm{H_2O,\,out}}/n_{\mathrm{fuel,in}})(MW_{\mathrm{H_2O,\,out}}/MW_{\mathrm{fuel,in}})$ (Eq. 7.5)
LHV = 23875 Btu/lb – (1055 Btu/lb)(2/1)(18/16) = **21500 Btu/lb**

Example 7.2 Estimating the heating values from the standard heat of combustion
The combustion reaction of 1 mol propane is:

$$C_3H_8(g) + 5O_2(g) \rightarrow 3CO_2(g) + 4H_2O(g)$$

Estimate the higher and lower heating values of 1 kg of propane at room temperature. Heat of vaporization is assumed as ΔH_{vap} = 2,442 kJ/kg (25°C).

Solution:
When the products and reactants are at their standard states (ideal-gas state at 1 bar and 25°C) the standard heat of reaction is expressed by

$$\Delta H_r^o = \sum_i v_i \Delta H_{fi}^o = v_{CO_2}\Delta H_{fCO_2}^o + v_{H_2O}\Delta H_{fH_2O}^o + v_{C_3H_8}\Delta H_{fC_3H_8}^o$$

Stoichiometric coefficients: $v_{CO_2} = 3$, $v_{H_2O} = 4$, and $v_{C_3H_8} = -1$.
Using the standard heats of formation form Table C1, the heat of combustion of one mole of propane (C_3H_8) is estimated by

$$\begin{aligned}\Delta H_{r298}^o &= 3(-393.51)\,\mathrm{kJ/gmol} + 4(-241.818)\,\mathrm{kJ/gmol} - (-104.7)\,\mathrm{kJ/gmol}\\ &= -2043.1\,\mathrm{kJ/gmol}\end{aligned}$$

Here the standard heat of reaction for oxygen is zero, as it is a naturally existing molecule in the environment.

Molecular weight of propane: MW = 44 g/gmol = 0.044 kg/mol. (Table A1)
Heat released after the combustion of 1 kg of propane is the lower heating value since the water product is at vapor state:

$$\Delta H_{r298}^o = \mathrm{LHV} = (2043.1\ \mathrm{kJ/gmol})(\mathrm{gmol}/0.044\ \mathrm{kg}) = 46{,}430\ \mathrm{kJ/kg}$$

The higher heating value:
MW_{H_2O} = (H_2) + 1/2(O_2) = 2 + 16 = 18 kg/kmol
$MW_{C_3H_8}$ = 3(C) + 8(H) = 36 + 8 = 44 kg/kmol
$n_{C_3H_8}$ = 1 gmol and n_{H2O} = 4 gmol
ΔH_{vap} = 2,442 kJ/kg (25°C)

After rearranging Eq. 7.5 higher heating value becomes:
HHV = LHV + (ΔH_{vap}) $(n_{H_2O,out}/n_{C_3H_8,in})(MW_{H_2O,out}/MW_{C_3H_8,in})$
HHV = 46,430 kJ/kg + (2,442 kJ/kg)(4/1)(18/44) = **50,426 kJ/kg**

7.4 Thermal Efficiency of Energy Conversions

Thermal efficiency is a measure of the amount of thermal energy that can be converted to another useful form. For an energy conversion device such as a boiler or furnace, the thermal efficiency η_{th} is the ratio of the amount of useful energy to the energy that went into the conversion

$$\eta_{th} = \frac{q_{useful}}{q_{in}} \tag{7.6}$$

No energy conversion device is 100% efficient. Most conversion devices and processes we use every day, such as light bulbs and steam power production operate with a low thermal efficiency. For example, most incandescent light bulbs are only 5–10% efficient because most of the electric energy is lost as heat to the

surroundings. An electric resistance heater has a thermal efficiency close to 100%. When comparing heating units, such as a highly efficient electric resistance heater with an 80% efficient natural gas-fueled furnace, an economic analysis is needed to determine the most cost-effective choice [10, 23, 33, 36].

Thermal efficiency usually involves a comparison of the total chemical energy in the fuels, and the useful energy extracted from the fuels in the form of kinetic energy. Which definition of heating value is being used as input energy affects the value of efficiency when the efficiency is determined by dividing the energy output by the input energy released by a fuel.

If a liquid water flow is used in a boiler to produce saturated vapor, the heat gained by the water can be expressed as the summation of sensible heat increase plus latent heat of vaporization

$$\dot{q} = \dot{m}(C_{p,av}\Delta T + \Delta H_{\text{vap}}) \tag{7.7}$$

where $\dot{q}$ is the heat gained, $\dot{m}$ is the mass flow rate, C_p is the specific heat capacity, ΔT is the temperature difference between inlet and outlet of the water in the boiler, and ΔH_{vap} is the heat of vaporization. Vaporization process needs heat and the condensation process releases heat as shown below

$$H_2O(\text{liquid}) + \Delta H_{\text{vap}} \rightarrow H_2O(\text{vapor}) \qquad \text{Vaporization} \tag{7.8}$$

$$H_2O(\text{vapor}) \rightarrow H_2O(\text{liquid}) + \Delta H_{\text{cond}} \qquad \text{Condensation} \tag{7.9}$$

The heat of vaporization is equal to the heat of condensation at the same temperature and pressure

$$\Delta H_{\text{vap}} = -\Delta H_{\text{cond}} \text{ (At the same temperature and pressure)} \tag{7.10}$$

The flue gases from boilers are in general not condensed. Therefore, actual amount of heat available to the boiler is the lower heating value since part of the heat of combustion of the fuel is used for the evaporation of the water. An accurate control of the air supply is essential to the boilers efficiency. Too much air cools the furnace and carries away useful heat, while the combustion will be incomplete with little air and unburned fuel will be carried over and smoke produced. *Net calorific value* of a fuel excludes the energy in the water vapor discharged to the stack in the combustion process. For heating systems their peak steady-state thermal efficiency is often stated as, for example, this furnace is 90% efficient, but a more detailed measure of seasonal energy effectiveness is the annual fuel utilization efficiency (AFUE), which is discussed in Sect. 9.4.

7.5 Ideal Fluid-Flow Energy Conversions

For fluid-flow systems, enthalpy rather than the internal energy is used in engineering since the fluid-flow work (PV) is taken into account within the enthalpy. Figure 7.2 shows an open steady-state flow system.

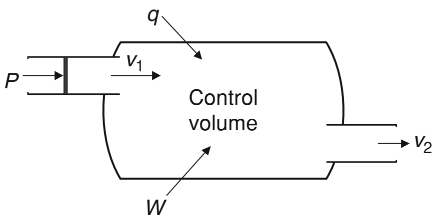


Fig. 7.2 Fluid-flow with heat and work interactions between the control volume and surroundings: $\dot{m}(\Delta H + \Delta v^2/2 + g\Delta z) = \dot{q} + \dot{W}_s$ at constant pressure

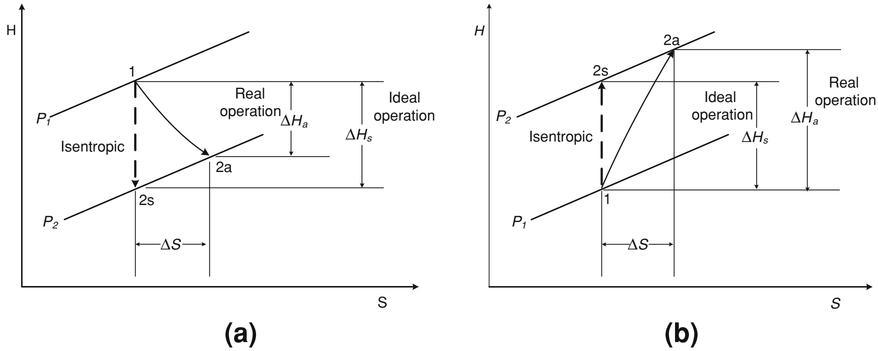


Fig. 7.3 Fluid flow work **a** in a turbine ($P_1 > P_2$), **b** in a compressor ($P_1 < P_2$); ideal operation is isentropic in a turbine and produces maximum work; ideal operation is isentropic in a compressor and requires minimum work

The general relation between the heat and work for a fluid-flow is expressed by:

$$\dot{m}\left(\Delta H + \frac{\Delta v^2}{2} + g\Delta z\right) = \dot{q} + \dot{W}_s \tag{7.11}$$

Heat transfer and work transfer should be distinguished based on the entropy transfer:

- Energy interaction that is accompanied by entropy transfer is heat transfer.
- Energy interaction that is not accompanied by entropy transfer is work.

For a process producing work, such a turbine as shown in Fig. 7.3a, the *ideal work* is the maximum possible work (isentropic) produced. However, for a process requiring work, such as a compressor shown in Fig. 7.3b, the ideal work is the minimum amount of required work (isentropic). These limiting values of work occur when the change of state in the process is accomplished reversibly [5, 27, 37]. For such processes friction, heat loss, and other losses are negligible. When the changes in kinetic and potential energies are negligible Eq. 7.11 becomes

$$\dot{m}\Delta H = \dot{q} + \dot{W}_s \tag{7.12}$$

For a uniform surrounding temperature T_o and using the definition of heat flow as

$$\dot{q} = T_o \Delta(\dot{m}S) \tag{7.13}$$

ideal work becomes

$$\dot{W}_{\text{ideal}} = \dot{W}_{\text{rev}} = \Delta(\dot{m}H) - T_o \Delta(\dot{m}S) \tag{7.14}$$

A reversible process, however, is hypothetical and used mainly for determination of the ideal work limit and comparing it with the actual work for the same property change. When the ideal work is produced (Fig. 7.3a), for example in a turbine, we have the *adiabatic or isentropic efficiency* defined by

$$\eta_{Ts} = \frac{W_{\text{prod}}}{W_{\text{ideal}}} = \frac{H_1 - H_{2a}}{H_1 - H_{2s}} = \frac{\Delta H_a}{\Delta H_s} \text{ (work produced)} \tag{7.15}$$

With the ideal work required (Fig. 7.3b), such as in a compressor, the *isentropic efficiency* η_s is obtained by

$$\eta_{Cs} = \frac{W_{\text{ideal}}}{W_{\text{req}}} = \frac{H_{2s} - H_1}{H_{2a} - H_1} = \frac{\Delta H_s}{\Delta H_a} \text{ (work required)} \tag{7.16}$$

Reversible work $\dot{W}_{\text{rev}}$ in terms of the values of exergy [20] for a process between the specified initial and final states is defined by

$$\dot{W}_{\text{rev}} = \dot{m}(Ex_1 - Ex_2) + \left(1 + \frac{T_o}{T}\right)\dot{q} \tag{7.17}$$

where T_o is the reference temperature. Example 7.3 illustrates the maximum expansion work calculations, while Example 7.4 calculates the isentropic efficiency.

Example 7.3 Maximum work (ideal work) calculations

One mole of air expands from an initial state of 500 K and 10 atm to the ambient conditions of the surroundings at 300 K and 1 atm. An average heat capacity of the air is $C_{p,av}$ = 29.5 J/mol K. Estimate the maximum work.

Solution:

Assume: air is ideal-gas system and the change of state is completely reversible.

T_1 = 500 K, P_1 = 10 atm, T_2 = 300 K, P_2 = 1 atm, $C_{p,av}$ = 29.5 J/mol K,
R = 8.314 J/mol K

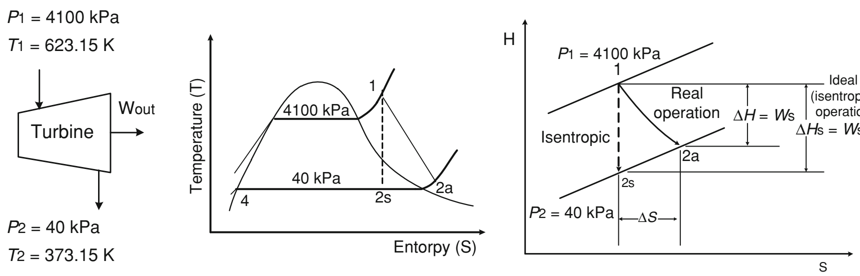
For an ideal-gas, enthalpy is independent of pressure and its change is

$$\Delta H = \int_1^2 C_{p,av} \mathrm{d}T = C_{p,av}(T_2 - T_1) = -5,900\,\mathrm{J/mol}$$

Change in entropy: $\Delta S = C_{p,av} \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) = 34.21\ \text{J/mol K},$ Eq. (4.50)

Ideal work: $W_{\text{ideal}} = W_{\text{rev}} = \Delta H - T_o\Delta S = \mathbf{-16163\ J/mol}$
The negative sign indicates that the expansion work is transferred to the surroundings.

Example 7.4 Isentropic turbine efficiency
An adiabatic turbine is used to produce electricity by expanding a superheated steam at 4,100 kPa and 350°C. The steam leaves the turbine at 40 kPa and 100°C. The steam mass flow rate is 8 kg/s. Determine the isentropic efficiency of the turbine.



Solution:
Assume: steady-state adiabatic operation. The changes in kinetic and potential energies are negligible.

Inlet conditions: superheated steam (Table F4)
P_1 = 4,100 kPa, T_1 = 623.15 K, H_1 = 3,092.8 kJ/kg, S_1 = 6.5727 kJ/kg K
Exit conditions: saturated steam (Table F3)
P_2 = 40 kPa, T_2 = 373.15 K, H_{2a} = 2,683.8 kJ/kg,
$S_{2\text{sat vap}}$ = 7.6709 kJ/kg K, $S_{2\text{sat liq}}$ = 1.2026 kJ/kg K at 40 kPa
$H_{2\text{sat vap}}$ = 2,636.9 kJ/kg, $H_{2\text{sat liq}}$ = 317.6 kJ/kg at 40 kPa (Table F4)
For the isentropic operation $S_1 = S_2$ = 6.5727 kJ/kg K
Since $S_{2\text{sat liq}} < S_2 = S_{2\text{sat vap}}$ the steam at the exit is saturated liquid–vapor mixture, and the quality of that mixture x_{2s} is

$$x_{2s} = \frac{S_2 - S_{2\text{sat liq}}}{S_{2\text{sat vap}} - S_{2\text{sat liq}}} = \frac{6.5727 - 1.2026}{7.6709 - 1.2026} = 0.83$$

$$H_{2s} = (1 - x_{2s})H_{2\text{sat liq}} + x_{2s}H_{2\text{sat vap}} = 2{,}243.1\ \text{kJ/kg (Isentropic)}$$

H_{2a} = 2,683.8 kJ/kg (Real operation)
H_1 = 3,092.8 kJ/kg

Isentropic efficiency becomes: $\eta_{Ts} = \frac{H_1 - H_{2a}}{H_1 - H_{2s}} = \mathbf{0.48}$ **(or 48%)**

7.6 Lost Work

The difference between the ideal work and the actual work is due to irreversibilities within the selected path between the initial and final states for a process. The extent of irreversibility is equivalent to exergy lost and is a measure of lost work potential [7]. Lost work is defined as a difference between the actual and ideal work and related to the rate of entropy production. Actual work is

$$\dot{W}_{\text{act}} = \Delta\left[\dot{m}\left(H + \frac{v^2}{2} + gz\right)\right] - \dot{q} \tag{7.18}$$

For surrounding temperature T_o, ideal work is

$$\dot{W}_{\text{ideal}} = \Delta\left[\dot{m}\left(H + \frac{v^2}{2} + gz\right)\right] - T_o\Delta(\dot{m}S) \tag{7.19}$$

Then the lost work

$$\dot{W}_{\text{lost}} = \dot{W}_{\text{act}} - \dot{W}_{\text{ideal}} = \dot{q} - T_o\Delta(\dot{m}S) \tag{7.20}$$

The lost work occurs because of the rate entropy production $\dot{S}_{\text{prod}}$, which is defined from the entropy balance in Sect. 5.4 by Eq. 5.20

$$\dot{S}_{\text{prod}} = \Delta(\dot{m}S) - \frac{\dot{q}}{T_o} \tag{7.21}$$

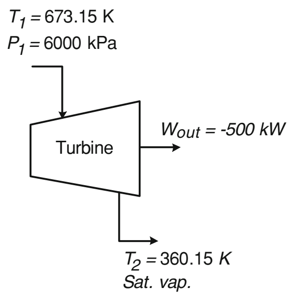
Therefore Eq. (7.20) becomes

$$\dot{W}_{\text{lost}} = T_o\dot{S}_{\text{prod}} \tag{7.22}$$

Since the rate of entropy production is always positive $(\dot{S}_{\text{prod}} > 0)$ for irreversible processes, then $\dot{W}_{\text{lost}} > 0$. Only when a process is completely reversible then $\dot{W}_{\text{lost}} = 0$ since $\dot{S}_{\text{prod}} = 0$. The rate of entropy production and hence the amount of lost work will increase as the irreversibility increases for a process. This leads to increase in the amount of energy that is unavailable and hence wasted. Example 7.5 illustrates the lost work calculations, while 7.6 estimates the minimum power required in a compressor.

Example 7.5 Estimation of lost work

A turbine discharges steam from 6 MPa and 400°C to saturated vapor at 360.15 K while producing 500 kW of shaft work. The temperature of surroundings is 290 K. Determine maximum possible production of power in kW and the amount of work lost.



Solution:
Assume: the turbine operates at steady-state. Kinetic and potential energy changes are negligible.

Basis: 1 kg/s steam flow rate
Inlet: superheated steam: H_1 = 3,180.1 kJ/kg, S_1 = 6.5462 kJ/kg K (Table F4)
Outlet: saturated steam: H_2 = 2,655.3 kJ/kg, S_2 = 7.5189 kJ/kg (Table F3)
W_{out} = − 500 kW, T_o = 290 K
$\Delta H = H_2 - H_1$ = (2,655.3 − 3,180.1) kJ/kg = − 524.8 kJ/kg
The amount of heat transfer: $q = -W + H_2 - H_1 = -24.8$ kJ/kg
We can determine the entropy production from an entropy balance on the turbine operating at steady-state that exchanges heat only with the surroundings:

$$S_{\text{prod}} = S_2 - S_1 - \frac{q}{T_o} = 7.5189 - 6.5462 + 24.8\ \text{kJ/kg}/(290)\text{K} = 1.06\ \text{kJ/kg K}$$

Lost work: $W_{\text{lost}} = T_o S_{\text{prod}} = 307.4$ kJ/kg
Maximum work output:

$$W_{\text{ideal}} = W_{\text{max}} = W_{\text{s}} - T_o S_{\text{prod}} = -500 - 307.4 = \mathbf{-807.4\,kJ/kg}$$

Example 7.6 Estimation of a minimum power required in a compressor
A compressor receives air at 15 psia and 80°F with a flow rate of 1.0 lb/s. The air exits at 40 psia and 300°F. Estimate the minimum power input to the compressor. The surroundings are at 520 R.

Solution:
Assume that potential energy effects are negligible, and steady process. Adiabatic compression with q_{loss} = 0.

Basis: air flow rate = $\dot{m}$ = 1 lb/s. The surroundings are at 520 R
The properties of air from Table D1 after conversions from SI units:

State 1: P_1 = 15 psia, T_1 = 540 R, H_1 = 129.0 Btu/lb, S_1 = 0.6008 Btu/lb R

State 2: P_2 = 40 psia, T_2 = 760 R, H_2 = 182.0 Btu/lb, S_2 = 0.6831 Btu/lb R
Compressor work:
$\dot{W}_s = \dot{m}(H_2 - H_1) = (1\ \text{lb/s})(182.08 - 129.06)\,\text{Btu/lb} = 53.0\ \text{Btu/s}$

The entropy production: $\dot{S}_{\text{prod}} = \dot{m}(S_2 - S_1) = 0.0823\ \text{Btu/s R}$ (Eq. 7.21)
Lost work: $\dot{W}_{\text{lost}} = \dot{m}T_o S_{\text{prod}} = 42.8\ \text{Btu/s}$
Minimum work required:
$\dot{W}_{\text{ideal}} = \dot{W}_{\min} = \dot{m}\left(\Delta H - T_o S_{\text{prod}}\right) = (53.0 - 42.8)\ \text{Btu/s} = \mathbf{10.2\,Btu/s}$

7.7 Efficiency of Mechanical Conversions

Transfer of mechanical energy is usually accomplished through a rotating shaft. When there is no loss (for example, in the form of friction) mechanical energy can be converted completely from one mechanical form to another. The mechanical energy conversion efficiency is estimated by

$$\eta_{mech} = \frac{\text{energy out}}{\text{energy in}} \tag{7.23}$$

For example, a mechanical energy conversion efficiency of 95% shows that 5% of the mechanical energy is converted to heat as a result of friction and other losses [6, 10].

In fluid-flow systems, a pump receives shaft work usually from an electric motor, and transfers this shaft work partly to the fluid as mechanical energy and partly to the frictional losses as heat. As a result, the fluid pressure or velocity, or elevation is increased. A turbine, however, converts the mechanical energy of a fluid to shaft work. The energy efficiency for a pump is defined by

$$\eta_{pump} = \frac{\text{mechanical energy out}}{\text{mechanical energy in}} = \frac{\dot{E}_{\text{mech out}}}{\dot{W}_{\text{shaft in}}} \tag{7.24}$$

Mechanical energy efficiency of a fan is the ratio of kinetic energy of air at the exit to the mechanical power input. If a fan, for example, is using a 50 W motor and producing an air flow velocity of 15 m/s and the air mass flow rate of 0.3 kg/s, then the mechanical energy efficiency of the fan becomes

$$\eta_{fan} = \frac{\dot{E}_{\text{mech out}}}{\dot{W}_{\text{shaft in}}} = \frac{(0.3\,\text{kg/s})(15\,\text{m/s})^2/2}{50\ \text{W}} = 0.675$$

Here, the velocity at the inlet is zero, v_1 = 0, and the pressure energy and potential energy are zero (ΔP = 0, and Δz = 0). The energy efficiency for a turbine is defined by

$$\eta_{turb} = \frac{\text{mechanical energy output}}{\text{mechanical energy (extracted from fluid) in}} = \frac{\dot{W}_{\text{shaft out}}}{\dot{E}_{\text{mech in}}} \tag{7.25}$$

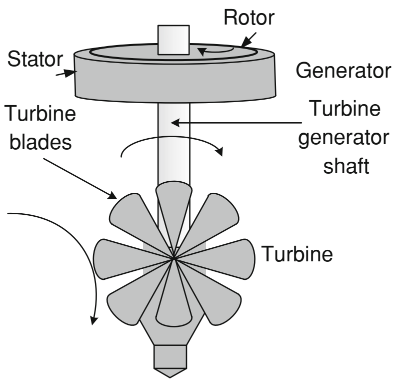


Fig. 7.4 A typical turbine and generator. The overall efficiency of a hydraulic turbine-generator is the ratio of the thermal energy of the water converted to the electrical energy; for a turbine efficiency of 0.8 and a generator efficiency of 0.9, we have: $\eta_{\text{turb-gen}} = \eta_{\text{turb}}\eta_{\text{gen}} = (0.8)(0.95) = 0.76$

The motor efficiency and the generator efficiency, on the other hand, are defined by

$$\eta_{motor} = \frac{\text{mechanical power output}}{\text{electrical energy input}} = \frac{\dot{W}_{\text{shaft out}}}{\dot{W}_{\text{elect in}}} \quad \text{(motor)} \tag{7.26}$$

$$\eta_{gen} = \frac{\text{electrical power output}}{\text{mechanical power input}} = \frac{\dot{W}_{\text{elect out}}}{\dot{W}_{\text{shaft in}}} \quad \text{(generator)} \tag{7.27}$$

A hydraulic turbine is combined with its generator in power production cycles, as shown in Fig. 7.4. A pump is also combined with its motor. Therefore, a combined or overall efficiency for turbine-generator and pump-motor systems is defined as follows

$$\eta_{\text{turb-gen}} = \eta_{\text{turb}}\eta_{\text{gen}} = \frac{\dot{W}_{\text{electout}}}{\Delta\dot{E}_{\text{mechin}}} \quad \text{(turbine-generator systems)} \tag{7.28}$$

The overall efficiency of a hydraulic turbine-generator shows the fraction of the mechanical energy of the water converted to electrical energy.

$$\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}} = \frac{\Delta\dot{E}_{\text{mech}}}{\dot{W}_{\text{electin}}} \quad \text{(pump-motor systems)} \tag{7.29}$$

The overall efficiency of a pump-motor system shows the fraction of the electrical energy converted to mechanical energy of the fluid [5, 37]. Example 7.7 illustrates the estimation of heat loss in an electric motor, while Example 7.8 illustrates the estimation of mechanical efficiency of a pump.

Example 7.7 Heat loss in an electric motor

An electric motor attached to a pump draws 10.0 A at 110 V. At steady load the motor delivers 1.32 hp of mechanical energy. Estimate the heat loss from the motor.

Solution:

Assume that the load to motor is steady.

$I = 10.0$ A, $V = 110$ V, $\dot{W}_s = 1.32$ hp $= 983$ W
The electric power received by the motor is used to create a pump work and heat:

Power received: $\dot{W}_e = IV = 1{,}100\,\text{W}$
Heat loss: $\dot{q}_{\text{loss}} = \dot{W}_e - \dot{W}_s = 1{,}100 - 983 = \mathbf{117\,W}$.
Only 983 kW of the 1,100 kW is converted to the pump work.

Example 7.8 Mechanical efficiency of a pump
The pump of a water storage tank is powered with a 16-kW electric motor operating with an efficiency of 90%. The water flow rate is 55 l/s. The diameters of the inlet and exit pipes are the same, and the elevation difference between the inlet and outlet is negligible. The absolute pressures at the inlet and outlet are 100 and 300 kPa, respectively. Determine the mechanical efficiency of the pump.

Solution:
Assume: steady-state one-dimensional and adiabatic flow. Kinetic and potential energy changes are negligible.
$\rho_{\text{water}} = 1{,}000\ \text{kg/m}^3 = 1\ \text{kg/l}$
Mass flow rate of water: $\dot{m}_{\text{water}} = \rho\dot{Q} = (1\ \text{kg/l})(55\ \text{l/s}) = 55\ \text{kg/s}$
Electric motor delivers the mechanical shaft work:
$W_{\text{pump}} = \eta_{\text{motor}}\dot{W}_{\text{electric}} = (0.9)(16\ \text{kW}) = 14.4\ \text{kW}$
Increase in the mechanical energy of the water:
$$\Delta\dot{E}_{\text{mech}} = \dot{m}\left(\frac{P_2 - P_1}{\rho} + \frac{v_2^2 - v_1^2}{2} + g(z_2 - z_1)\right)$$
After neglecting kinetic and potential energies:
$$\Delta\dot{E}_{\text{mech}} = \dot{m}\left(\frac{P_2 - P_1}{\rho}\right) = (55\,\text{kg/s})\left(\frac{(300-100)\text{kPa}}{1000\ \text{kg/m}^3}\right)\left(\frac{\text{kJ}}{\text{kPa m}^3}\right) = 11.0\,\text{kW}$$
The mechanical efficiency of the pump is
$$\eta_{\text{pump}} = \frac{\Delta\dot{E}_{\text{mech}}}{\dot{W}_{\text{pump}}} = \frac{11\,\text{kW}}{14.4\ \text{kW}} = \mathbf{0.763\ or\ 76.3\%}$$
Only 11 kW of the 14.4 kW received by the pump is converted to the pump work. Remaining 3.4 kW is lost as heat because of friction.

7.8 Conversion of Thermal Energy by Heat Engines

Heat engines transform thermal energy q_{in} into mechanical energy or work W_{out}. Some examples of heat engines include the steam engine, heat pump, gasoline (petrol) engine in an automobile, the diesel engine, and gas power cycles. All of

Table 7.2 Comparison of the processes for some of the heat engines

Cycle	Compression	Heat addition	Expansion	Heat rejection
Rankine	Adiabatic	Isobaric	Adiabatic	Isobaric
Carnot	Isentropic	Isothermal	Isentropic	Isothermal
Ericsson	Isothermal	Isobaric	Isothermal	Isobaric
Stirling	Isothermal	Isochoric	Isothermal	Isochoric
Diesel	Adiabatic	Isobaric	Adiabatic	Isochoric
Otto	Adiabatic	Isochoric	Adiabatic	Isochoric
Brayton	Adiabatic	Isobaric	Adiabatic	Isobaric

Table 7.3 Various types of cycles for heat engines

External combustion cycles	Without phase change: Brayton, Carnot, Ericsson, Stirling With phase change: Rankine, Two-phased Stirling
Internal combustion cycles	Without phase change: Diesel, Otto
Refrigeration cycles	Vapor-compression
Heat pump cycles	Vapor-compression

these heat engines drive the mechanical motion of the engine by the expansion of heated gases. Some heat engines operate with phase-change cycles, such as steam power production by Rankine cycle where liquid water changes to vapor after adding heat. After expansion in the cycle, the vapor condenses into liquid water. Other type of heat engines operate without phase-change cycles, such as Brayton cycle where a hot gas is cooled after the expansion in the heat engine. Typical gas power cycles consists of compressing cool gas, heating the gas, expanding the hot gas, and finally cooling the gas before repeating the cycle. Table 7.2 lists some important engines and their cycle processes. Typical thermal efficiency for power plants in the industry is around 33% for coal- and oil-fired plants, and up to 50% for combined-cycle gas-fired plants. Combined-cycle gas-turbine plants are driven by both steam and natural gas [13, 15, 28].
Each process in a cycle is at one of the following states:

- Isothermal (at constant temperature)
- Isobaric (at constant pressure)
- Isochoric (at constant-volume)
- Adiabatic (no heat is added or removed)

Engine types vary as shown in Table 7.3. In the combustion cycles, once the mixture of air and fuel is ignited and burnt, the available energy can be transformed into the work by the engine. In a reciprocating engine, the high-pressure gases inside the cylinders drive the engine's pistons. Once the available energy has been converted to mechanical work, the remaining hot gases are vented out and this allows the pistons to return to their previous positions [15].

Some of the available heat input is not converted into work, but is dissipated as waste heat q_{out} into the environment. The thermal efficiency of a heat engine is the percentage of heat energy that is converted into work, and is estimated by

$$\eta_{th} = \frac{W_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \tag{7.30}$$

The efficiency of even the best heat engines is low; usually below 50% and often far below. So the energy lost to the environment by heat engines is a major waste of energy resources, although modern cogeneration, combined cycle, and energy recycling schemes are beginning to use this waste heat for other purposes. This inefficiency can be attributed mainly to three causes:

- Thermal efficiency of any heat engine is limited by the Carnot efficiency.
- Specific types of engines may have lower limits on their efficiency due to the inherent irreversibility of the engine cycle they use.
- The non-ideal behavior of real engines, such as mechanical friction and losses in the combustion process, may cause further efficiency losses.

Thermal efficiency of heat engine cycles cannot exceed the limit defined by the Carnot cycle which states that the overall efficiency is dictated by the difference between the lower and upper operating temperatures of the engine. This limit assumes that the engine is operating in ideal conditions that are frictionless processes, ideal-gases, perfect insulators, and operation at infinite time. A car engine's real-world fuel economy is usually measured in the units of miles per gallon (or fuel consumption in liters per 100 km). Even when aided with turbochargers and design aids, most engines retain an *average* efficiency of about 18–20%. Rocket engine efficiencies are still better, up to 70%, because they combust at very high temperatures and pressures, and are able to have very high expansion ratios. For stationary and shaft engines, fuel consumption is measured by calculating the *brake specific fuel consumption* which measures the mass flow rate of fuel consumed divided by the power produced. Example 7.8 illustrates the estimation of fuel consumption of a car.

The engine efficiency alone is only one factor. For a more meaningful comparison, the overall efficiency of the entire energy supply chain from the fuel source to the consumer should be considered. Although the heat wasted by heat engines is usually the largest source of inefficiency, factors such as the energy cost of fuel refining and transportation, and energy loss in electrical transmission lines may offset the advantage of a more efficient heat engine. Engines must be optimized for other goals besides efficiency such as low pollution. Vehicle engines must also be designed for low emissions, adequate acceleration, fast starting, light weight, and low noise. These requirements may lead to compromises in design that may reduce efficiency. Large stationary electric generating plants have fewer of these competing requirements so the Rankine cycles are significantly more efficient than vehicle engines. Therefore, replacing internal combustion vehicles with electric vehicles, which run on a battery recharged with electricity generated by

burning fuel in a power plant, has the theoretical potential to increase the thermal efficiency of energy use in transportation, thus decreasing the demand for fossil fuels [6, 29].

Real engines have many departures from ideal behavior that waste energy, reducing actual efficiencies far below the theoretical values given above. Examples of non-ideal behavior are:

- Friction of moving parts.
- Inefficient combustion.
- Heat loss from the combustion chamber.
- Non-ideal behavior of the working fluid.
- Inefficient compressors and turbines.
- Imperfect valve timing.

Example 7.9 Thermal efficiency of a heat engine

Heat is transferred to a heat engine from a furnace at a rate of 90 MW. The waste heat is discharged to the surroundings at a rate of 55 MW. Estimate the net power output and thermal efficiency of the engine if all the other power losses are neglected.

Solution:
Assume: steady-state process with negligible heat losses.

$\dot{q}_{in} = 90$ MW, $\dot{q}_{out} = 55$ MW

Net power output: $\dot{W}_{\text{net out}} = \dot{q}_{\text{in}} - \dot{q}_{\text{out}} = \mathbf{35\,MW}$

Thermal efficiency: $\eta_{th} = \dfrac{\dot{W}_{\text{net}}}{q_{\text{in}}} = \dfrac{35}{90} = \mathbf{0.388}$ **(or 38.8%)**

The heat engine can convert 38.8% of the heat transferred from the furnace.

Example 7.10 Fuel consumption of a car

The overall efficiencies are about 25–28% for gasoline car engines, 34–38% for diesel engines, and 40–60% for large power plants [6]. A car engine with a power output of 120 hp has a thermal efficiency of 24%. Determine the fuel consumption of the car if the fuel has a higher heating value of 20,400 Btu/lb.

Solution:
Assume: the car has a constant power output.
Net heating value ≅ higher heating value (0.9) = 18,360 Btu/lb (Approximate)
Car engine power output and efficiency: $\dot{W}_{\text{net}} = 120\,\text{hp}, \eta_{\text{th}} = 0.24$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{q}_{\text{in}}} \rightarrow \dot{q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{120\text{ hp}}{0.24}\left(\frac{2{,}545\text{Btu/h}}{\text{hp}}\right)$$

$\dot{q}_{in} = 1{,}272{,}500\,\text{Btu/hr}$

Net heating value ≅ higher heating value (0.9) = 18,360 Btu/lb

$$\text{Fuel consumption} = \dot{q}_{in}/\text{net heating value} = 1{,}272{,}500\,\text{Btu/h}/18{,}360\,\text{Btu/lb} = \mathbf{69.3\,lb/h}$$

Assuming an average density of 0.75 kg/l:
ρ_{gas} = (0.75 kg/l)(2.2 lb/kg)(l/0.264 gal) = 6.25 lb/gal
Fuel consumption in terms of gallon: (69.3 lb/h)/(6.25 lb/gal) = **11.1 gal/h**

7.8.1 Air-Standard Assumptions

Internal combustion cycles of Otto and Diesel engines as well as the gas turbines are some well-known examples of engines that operate on gas cycles. In the gas power cycles, the working fluid remains gas for the entire cycle. In the analysis of gas power cycles, the following assumptions known as *air-standard assumptions* are used:

- Working fluid is air and always behaves as an ideal gas.
- All processes in the cycle are internally reversible (isentropic).
- Heat-addition process uses an external heat source.
- Heat-rejection process restores the working fluid to its original state.

It is also assumed that the air has a constant value for the ratio of specific heats determined at room temperature (25°C or 77°F). With this air-standard assumptions are called the *cold-air-standard assumptions*. These assumptions simplify the analysis of gas power cycles without significantly deviating from the actual cycle [6].

7.8.2 Isentropic Processes of Ideal Gases

Entropy change of ideal gas is expressed by

$$\Delta S = S_2 - S_1 = \int_1^2 C_p(T)\frac{dT}{T} - R\ln\left(\frac{P_2}{P_1}\right) \tag{7.31}$$

Under isentropic conditions (ΔS = 0) and constant heat capacity $C_{p,av}$, Eq. 7.31 becomes

$$0 = C_{p,av}\ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \tag{7.32}$$

For ideal-gas, the heat capacities are related by

$$C_p - C_v = R \tag{7.33}$$

From Eqs. 7.32 and 7.33 the following relations may be derived

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} \text{ and } \left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} \tag{7.34}$$

where $\gamma = C_p/C_v$.

Under isentropic conditions (ΔS = 0) and variable heat capacities, Eq. 7.31 becomes

$$0 = S_{T2} - S_{T1} - R\ln\left(\frac{P_2}{P_1}\right) \tag{7.35}$$

where S_{T1} and S_{T2} are the values of entropy at temperatures T_1 and T_2, respectively. After rearranging, Eq. (7.35) yields,

$$\frac{P_2}{P_1} = \exp\left(\frac{S_{T2} - S_{T1}}{R}\right) = \frac{\exp(S_{T2}/R)}{\exp(S_{T1}/R)} = \frac{P_{r2}}{P_{r1}} \tag{7.36}$$

where P_{r1} is called the *relative pressure* given by exp(S_{T1}/R), which is dimensionless quantity. Using the ideal-gas relation

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{7.37}$$

the relative specific volume V_r ($V_r = T/P_r$) is derived by

$$\frac{V_2}{V_1} = \frac{T_2/P_{r2}}{T_1/P_{r1}} = \frac{V_{r2}}{V_{r1}} \tag{7.38}$$

The values of P_r and V_r are tabulated for the air against temperature in Appendix D, Table D1. Equations 7.35–7.38 account the variation of specific heats with temperature and are valid only for isentropic processes of ideal-gases. Equations 7.36 and 7.38 are useful in the analysis of gas power cycles operating with isentropic processes [5].

7.8.3 Conversion of Mechanical Energy by Electric Generator

In electricity generation, an *electric generator* converts mechanical energy to electrical energy generally using *electromagnetic induction*. Electromagnetic induction is the production of electric potential (voltage) across a conductor moving through a magnetic field. A generator forces electrons in the windings to flow through the external electrical circuit. The source of mechanical energy may be water falling through a turbine, waterwheel, an internal combustion engine, and a wind turbine. The reverse is the conversion of electrical energy into mechanical energy, which is done by an electric motor. Figure 7.4 shows a schematic view of a generator.

The efficiency of a generator is determined by the power of the load circuit and the total power produced by the generator. For most commercial electrical

generators, this ratio can be as high as of 95%. The losses typically arise from the transformer, the copper windings, magnetizing losses in the core, and the rotational friction of the generator. The overall efficiency of a hydraulic turbine-generator is the ratio of the thermal energy of the water converted to the electrical energy and obtained by

$$\eta_{\text{turb}-\text{gen}} = \eta_{\text{turb}}\eta_{\text{gen}} \tag{7.39}$$

Michael Faraday discovered the operating principle of electromagnetic generators. The principle, later called Faraday's law, is that an electromotive force is generated in an electrical conductor that encircles a varying magnetic flux. The two main parts of a generator are rotor and stator. Rotor is the rotating part, while stator is the stationary part. The armature of generator windings generates the electric current. The armature can be on either the rotor or the stator. Because power transferred into the field circuit is much less than in the armature circuit, alternating current generators mostly have the field winding on the rotor and the stator as the armature winding.

An electric generator or electric motor uses magnetic field coils. If the field coils are not powered, the rotor in a generator can spin without producing any usable electrical energy. When the generator first starts to turn, the small amount of magnetism present in the iron core provides a magnetic field to get it started, generating a small current in the armature. This flows through the field coils, creating a larger magnetic field which generates a larger armature current. This "bootstrap" process continues until the magnetic field in the core levels off due to saturation and the generator reaches a steady-state power output. Very large power station generators often utilize a separate smaller generator to excite the field coils of the larger generator [6].

The *dynamo* was the first electrical generator capable of delivering power for industry. The dynamo uses electromagnetic principles to convert mechanical rotation into the direct current. A dynamo consists of a stationary structure, which provides a constant magnetic field, and a set of rotating windings which turn within that field. A *commutator* is a rotary electrical switch in certain types of electric motors or electrical generators that periodically reverses the current direction between the rotor and the external circuit.

7.8.4 Carnot Engine Efficiency

A Carnot *heat engine* converts heat to mechanical energy by bringing a working fluid from a high temperature state T_H to a lower temperature state T_C. Figure 7.5 shows a typical pressure–volume *PV* and temperature-entropy *TS* diagrams of ideal engine cycles. On both the *PV* and *TS* diagrams, the area enclosed by the process curves of a cycle represents the net heat transfer to be converted to mechanical energy by the engine. Therefore, any modifications that improve the net heat transfer rate will also improve the thermal efficiency [37].

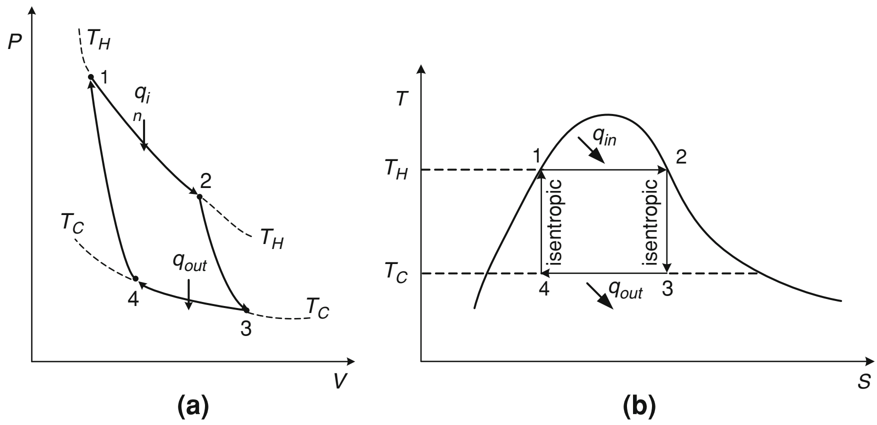


Fig. 7.5 Typical ideal engine cycles **a** on a pressure–volume *PV* diagram, **b** on a temperature-entropy *TS* diagram

A heat source heats the working fluid in the high temperature state. The working fluid generates work in the engine while transferring remaining heat to the colder sink until it reaches a low temperature state. The working fluid usually is a gas or liquid. During the operation of an engine some of the thermal energy is converted into work and the remaining energy is lost to a heat sink, mainly the general surroundings.

Carnot cycle is composed of four totally reversible processes shown in Fig. 7.5:

- Process 1–2: isothermal heat-addition at constant temperature T_H
- Process 2–3: isentropic expansion at constant entropy $S_2 = S_3$
- Process 3–4: isothermal heat rejection at constant temperature T_C
- Process 4–1: isentropic compression at constant entropy $S_4 = S_1$

Thermal efficiency of the Carnot engine is calculated by

$$\eta_{\mathrm{th}} = 1 - \frac{q_{\mathrm{out}}}{q_{\mathrm{in}}} \tag{7.40}$$

From Fig. 7.5b, we can estimate the amounts of added q_{in} and rejected heat q_{out} values as

$$q_{\mathrm{in}} = T_H(S_2 - S_1) \tag{7.41}$$

$$q_{\mathrm{out}} = T_C(S_4 - S_3) \tag{7.42}$$

Using Eqs. 7.41 and 7.42 in Eq. 7.40, we have

$$\eta_{\mathrm{th}} = 1 - \frac{T_C(S_4 - S_3)}{T_H(S_2 - S_1)} = 1 - \frac{T_C}{T_H} \tag{7.43}$$

since the power cycles are isentropic and we have $S_2 = S_3$ and $S_4 = S_1$. Equation 7.43 shows that heat engines efficiency is limited by Carnot's efficiency which is

equal to the temperature difference between the hot (T_H) and cold (T_C) divided by the temperature at the hot end, all expressed in absolute temperatures. The Carnot cycle is a standard against which the actual or ideal cycle can be compared. The Carnot cycle has the highest thermal efficiency of all heat engines operating between the same heat source temperature T_H and the same sink temperature T_C.

The cold side of any heat engine is close to the ambient temperature of the environment of around 300 K, such as a lake, a river, or the surrounding air. Therefore, most efforts to improve the thermodynamic efficiencies of various heat engines focus on increasing the temperature of the hot source within material limits. The highest value of T_H is limited by the maximum temperature that the components of the heat engine, such as pistons or turbine blades, can withstand.

The thermal efficiency of Carnot engine is independent of the type of the working fluid, or whether the cycle operates in a closed or steady-flow system. For example, if an automobile engine burns gasoline at a temperature of $T_H = 1{,}200$ K and the air at exhaust temperature is $T_C = 800$ K, then its maximum possible efficiency is:

$$\eta_{\text{Carnot}} = 1 - \frac{800\text{K}}{1{,}200\ \text{K}} = 0.33 \text{ or } 33\%$$

Totally reversible engine cycle is very difficult to achieve in reality as it requires very large heat exchangers and needs very long time. Practical engines have efficiencies far below the Carnot limit. For example, the average automobile engine is less than 35% efficient. As Carnot's theorem only applies to heat engines, devices that convert the fuel's energy directly into work without burning it, such as fuel cells, can exceed the Carnot efficiency. Actual cycles involve friction, pressure drops as well as the heat losses in various processes in the cycle; hence they operate at lower thermal efficiencies [2, 4, 13, 21].

7.8.5 Endoreversible Heat Engine Efficiency

A Carnot engine must operate at an infinitesimal temperature gradient, and therefore the Carnot efficiency assumes the infinitesimal limit. Endoreversible engine gives an upper bound on the energy that can be derived from a real process that is lower than that predicted by the Carnot cycle, and accommodates the heat loss occurring as heat is transferred. This model predicts how well real-world heat engines can perform [3, 16] by using the following relationship

$$\eta_{\max} = 1 - \sqrt{\frac{T_C}{T_H}} \tag{7.44}$$

Table 7.4 compares the efficiencies of engines operating on the Carnot cycle, endoreversible cycle, and actual cycle. As can be seen, the values of observed efficiencies of actual operations are close to those obtained from the endoreversible cycles.

Table 7.4 Efficiencies of engines operating on the Carnot cycle, endoreversible cycle, and actual cycle

Power plant	T_C (°C)	T_H (°C)	η (Carnot)	η (Endoreversible)	η (Observed)
Coal-fired power plant	25	565	0.64	0.40	0.36
Nuclear power plant	25	300	0.48	0.28	0.30
Geothermal power plant	80	250	0.33	0.18	0.16

Callen [3]

7.8.6 Rankine Engine Efficiency

The Rankine cycle is used in steam turbine power plants. A turbine converts the kinetic energy of a moving fluid to mechanical energy. The steam is forced against a series of blades mounted on a shaft connected to the generator. The generator, in turn, converts its mechanical energy to electrical energy based on the relationship between magnetism and electricity. Most of the world's electric power is produced with Rankine cycle. Since the cycle's working fluid changes from liquid to vapor and back to liquid water during the cycle, their efficiencies depend on the properties of water. The thermal efficiency of modern steam turbine plants with reheat cycles can reach 47%, and in combined-cycle plants it can approach 60%.

Figure 7.6 describes the main processes within the Rankine cycle. The main processes of the Rankine cycle are:

Process 4–1: the water is pumped from low to high pressure.

Process 1–2: the high-pressure liquid enters a boiler where it is heated at constant pressure by an external heat source to become a vapor.

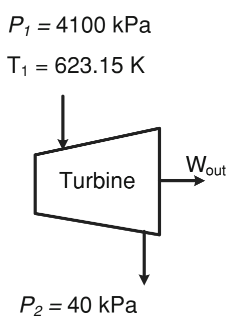
Process 2–3: the vapor expands through a turbine, generating power. This decreases the temperature and pressure of the vapor, and some condensation may occur.

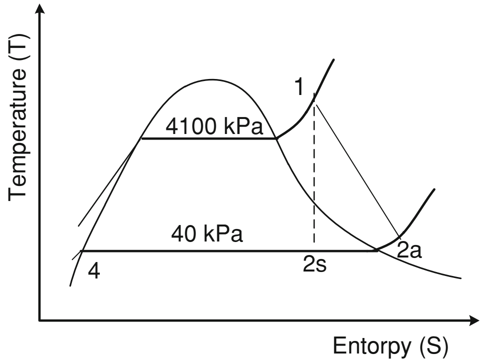
Process 3–4: the wet vapor then enters a condenser where it is condensed at a constant pressure to become a saturated liquid.

The Rankine cycle is sometimes referred to as a practical Carnot cycle (see Fig. 7.5) because in an ideal Rankine cycle the pump and turbine would be *isentropic* and produce no entropy and hence maximize the work output. The main difference is that heat addition (in the boiler) and rejection (in the condenser) are isobaric in the Rankine cycle and isentropic in the theoretical Carnot cycle [13, 37]. Example 7.1 illustrates the estimation of thermal efficiency of an ideal Rankine cycle, and Example 7.12 analyzes the power output of steam power plant.

Example 7.11 Steam turbine efficiency and power output

An adiabatic turbine is used to produce electricity by expanding a superheated steam at 4,100 kPa and 350°C. The steam leaves the turbine at 40 kPa. The steam mass flow rate is 8 kg/s. If the isentropic efficiency is 0.75, determine the actual power output of the turbine.





Solution:

Assume: steady-state adiabatic operation. The changes in kinetic and potential energies are negligible.

Data from Table F4:

Inlet conditions:

P_1 = 4,100 kPa, T_1 = 623.15 K, H_1 = 3,092.8 kJ/kg, S_1 = 6.5727 kJ/kg K

Exit conditions:

P_2 = 40 kPa, $S_{2\text{sat vap}}$ = 7.6709 kJ/kg K, $S_{2\text{sat liq}}$ = 1.0261 kJ/kg K

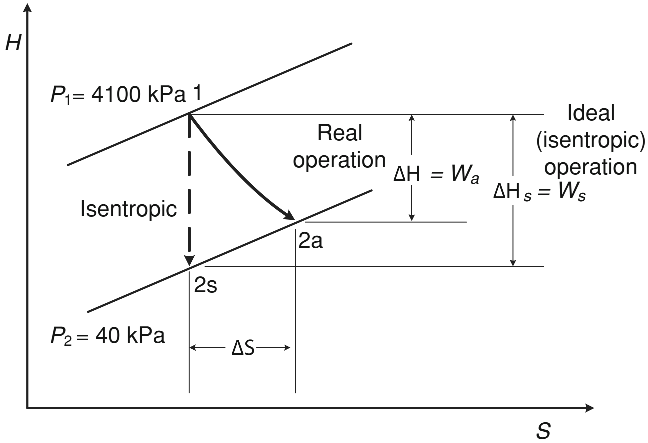
$H_{2\text{sat vap}}$ = 2,636.9 kJ/kg, $H_{2\text{sat liq}}$ = 317.6 kJ/kg at 40 kPa (Table F3)

For the isentropic operation $S_1 = S_2$ = 6.5727 kJ/kg K

Since $S_1 < S_{2\text{sat vap}}$ the steam at the exit is saturated liquid–vapor mixture

Quality of the saturated mixture: $x_{2s} = \dfrac{S_2 - S_{2\text{sat liq}}}{S_{2\text{sat vap}} - S_{2\text{sat liq}}} = \dfrac{6.5727 - 1.0261}{7.6709 - 1.0261} = 0.83$

$H_{2s} = (1 - x_{2s})H_{2\text{sat liq}} + x_{2s}H_{2\text{sat vap}} = 2243.1\,\text{kJ/kg}$



Using isentropic efficiency of 0.75, we can estimate the actual enthalpy H_{2a},

$$\eta_t = \frac{H_1 - H_{2a}}{H_1 - H_{2s}} = 0.75 \rightarrow H_{2a} = 2{,}455.5 \text{ kJ/kg}$$

Actual power output which is not isentropic $\dot{W}_a$ (see figure above)

$$\dot{m}\Delta H = \dot{W}_a = \dot{m}(H_{2a} - H_1) = (8 \text{ kg/s})(2{,}455.5 - 3{,}092.8) = -5{,}098.4 \text{ kW} \\ = \mathbf{-5.098\,MW}$$

Maximum power output: $\dot{W}_s$ (see figure above) is only achievable by isentropic expansion.

$$\dot{m}\Delta H = \dot{W}_a = \dot{m}(H_{2s} - H_1) = (8 \text{ kg/s})(2{,}243.1 - 3{,}092.8) = -6{,}797.6 \text{ kW} \\ = \mathbf{-6.797\,MW}$$

The Enthalpy-entropy *HS* diagram above explains the difference between real and ideal turbine operation. The signs are negative for work output.

Example 7.12 Estimation of thermal efficiency of a Rankine cycle
A Rankine cycle shown in Fig. 7.6 uses natural gas to produce 0.12 MW power. The combustion heat supplied to a boiler produces steam at 10,000 kPa and 798.15 K, which is sent to a turbine. The turbine efficiency is 0.7. The discharged steam from the turbine is at 30 kPa. The pump efficiency is 0.75. Determine:

(a) The thermal efficiency of an ideal Rankine cycle.
(b) The thermal efficiency of an actual Rankine cycle.

Solution:
Assume that kinetic and potential energy changes are negligible, and the system is at steady-state.
Consider Fig. 7.6.

(a) The basis is 1 kg/s steam.

Turbine in:

H_2 = 3,437.7 kJ/kg, S_2 = 6.6797 kJ/kg K, at T_2 = 798.15 K; P_2 = 10,000 kPa (Table F4)

Turbine out:

$H_{4\text{sat liq}} = H_{3\text{sat liq}}$ = 289.3 kJ/kg K; $H_{4\text{sat vap}} = H_{3\text{sat vap}}$ = 2,625.4 kJ/kg K;

at $P_3 = P_4$ = 30 kPa, $S_{3\text{sat liq}}$ = 0.9441 kJ/kg K; $S_{3\text{sat vap}}$ = 7.7695 kJ/kg K; at P_3 = 30 kPa (Table F3)

η_{turb} = 0.70; η_{pump} = 0.75; V = 1,020 cm^3/kg at T = 342.15 K

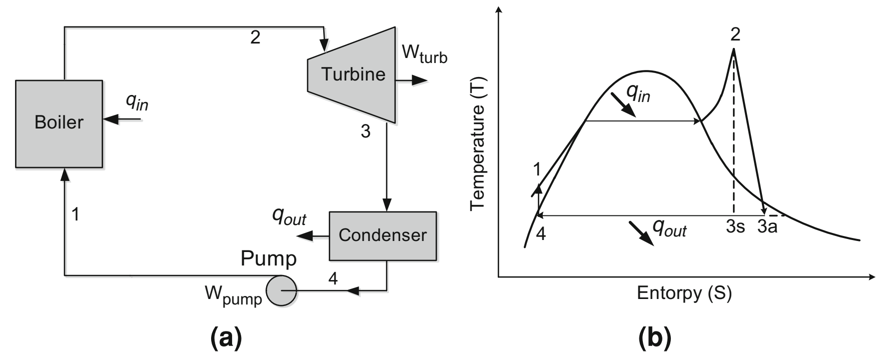
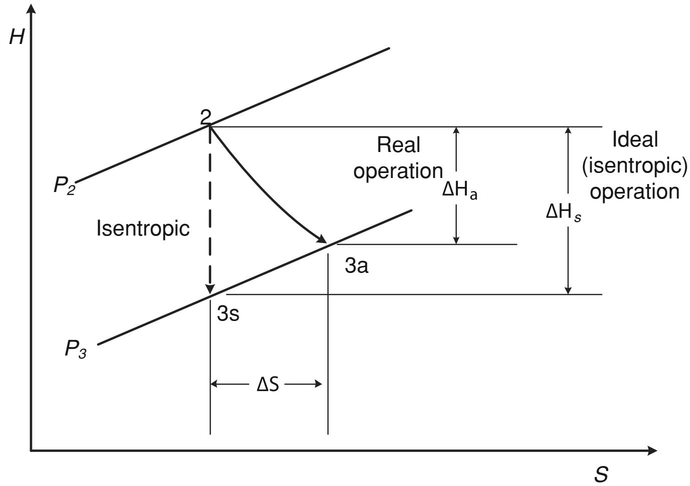


Fig. 7.6 **a** Schematic of Rankine cycle, **b** Rankine cycle processes on a T versus S diagram; Process 2–3s shows isentropic expansion, while process 2–3a shows a real expansion process where entropy is not constant and the amount of waste heat increases

For the ideal Rankine cycle, the operation is isentropic, and $S_2 = S_3$. However, $S_2 < S_{3\text{sat vap}}$, and the discharged steam from the turbine is wet steam.

The quality of the wet steam x_{3s}: $x_{3s} = \dfrac{S_2 - S_{3\text{sat liq}}}{S_{3\text{sat vap}} - S_{3\text{sat liq}}} = 0.84$

The enthalpy of the wet steam H_{3s}:
$H_{3s} = (1 - x_{3s})H_{3\text{ sat liq}} + x_{3s}H_{3\text{ sat liq}} = 2{,}252.4\,\text{kJ/kg}$
Ideal operation: $W_s = \Delta H_s = H_{3s} - H_2 = -1{,}185.3$ kJ/kg (isentropic expansion)



$P_3 = P_4$, $P_1 = P_2$
From the isentropic pump operation, we have: $W_{ps} = V(P_2 - P_3) = 10.2\,\text{kJ/kg}$
So the enthalpy at point 1: $H_{1s} = \Delta W_{ps} + H_4 = 10.2 + 289.3 = 299.5$ kJ/kg
The heat required in the boiler becomes: $q_{\text{in}} = H_2 - H_{1s} = 3138.2\,\text{kJ/kg}$
The net work for the ideal Rankine cycle:

$W_{\text{net},s} = W_p + \Delta H_s = (10.2 - 1185.3)\ \text{kJ/kg} = -1175.1\ \text{kJ/kg}$

The efficiency of the ideal Rankine cycle: $\eta_{\text{ideal cycle}} = \dfrac{W_{\text{net},s}}{q_{in}} = \mathbf{0.374\ or\ 37.4\%}$

(b) Actual cycle efficiency: $\dfrac{\Delta H_a}{\Delta H_s} = \eta_{t,\text{actual}}$

With the turbine efficiency of $\eta_t = 0.7$, we have: $\Delta H_a = \eta_t \Delta H_s = -829.7$ kJ/kg

Pump efficiency: $\dfrac{\Delta W_{ps}}{\Delta W_{pa}} = \eta_{\text{pump}}$: we have: $W_{pa} = \dfrac{W_{\text{ps}}}{\eta_{\text{pump}}} = 13.6\ \text{kJ/kg}$

The net work for the actual cycle: $W_{\text{net,act}} = W_{pa} + \Delta H_a = -816.1$ kJ/kg

$H_{1a} = \Delta W_{pa} + H_4 = (13.6 + 289.3)\ \text{kJ/kg} = 302.9\ \text{kJ/kg}$

$q_{\text{in,act}} = H_2 - H_{1a} = (3{,}437.7 - 302.9)\ \text{kJ/kg} = 3{,}134.81\ \text{kJ/kg}$

Therefore, the efficiency of the actual cycle: $\eta_{\text{actual}} = \dfrac{W_{\text{net,act}}}{q_{\text{in,act}}} = \mathbf{0.260\ or\ 26\%}$

Comparison of the two efficiencies shows that both operations have relatively low efficiencies, although the actual cycle is considerably less efficient than the ideal Rankine cycle (37.4%). The actual cycles involves friction and pressure drops as well as the heat losses in various processes; therefore have lower thermal efficiency than those of ideal cycles.

7.8.7 Brayton Engine Efficiency

The Brayton cycle was first proposed by George Brayton in around 1870. The Brayton cycle is used for gas turbines operating on an open cycle as shown in Fig. 7.7. The Brayton engine consists of three main components: a compressor, a combustion chamber, and a turbine. The air after being compressed in the compressor is heated by burning fuel in it. The heated air expands in a turbine and produces the power. The two main applications of gas-turbine engines are jet engines and electric power production. Jet engines take a large volume of hot gas from a combustion process and feed it through a nozzle which accelerates the plane to high speed (see Fig. 7.8). Gas-turbine cycle engines employ a continuous combustion system where compression, combustion, and expansion occur simultaneously at different places in the engine [32]. The combustion takes place at constant pressure. The fuel must be transportable to the combustion chamber, and that the fuel releases sufficient heat of combustion to produce necessary power.

Some of the power produced in the turbine of a gas-turbine power plant is used to drive the compressor. The ratio of the compressor work to the turbine work is called the *back work ratio*. Sometimes, more than one-half of the turbine work may be used by the compressor [21].

The Brayton cycle is analyzed as open system. By neglecting the changes in kinetic and potential energies, the energy balance on a unit-mass basis is

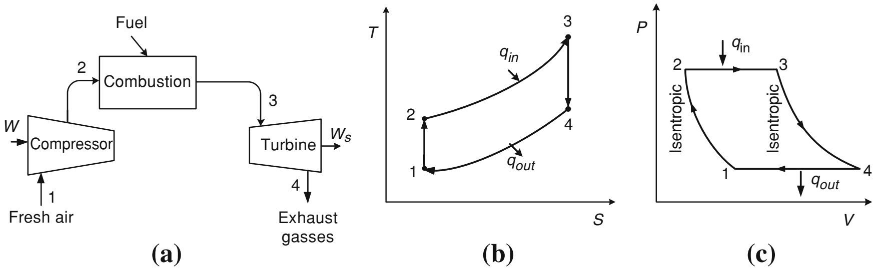


Fig. 7.7 **a** Schematic of open Brayton cycle, **b** ideal Brayton cycle on a *TS* diagram with processes of isentropic adiabatic compression, isobaric heat addition, isentropic adiabatic expansion, and isobaric heat rejection, **c** ideal Brayton cycle on a *PV* diagram

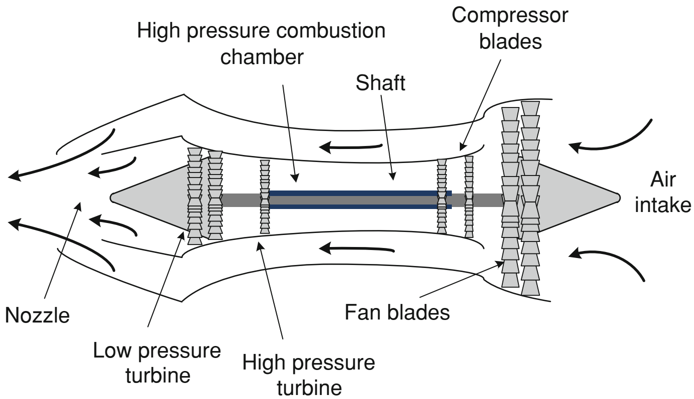


Fig. 7.8 Schematic of a jet engine using gas power cycle

$$\Delta H = (q_{\text{in}} - q_{\text{out}}) + (W_{\text{in}} - W_{\text{out}}) \tag{7.45}$$

Assuming constant heat capacity, thermal efficiency can be derived using the heat added q_{in} and heat rejected q_{out}

$$\eta_{\text{Brayton}} = 1 - \left(\frac{q_{\text{out}}}{q_{\text{in}}}\right) = 1 - \left(\frac{C_{p,av}(T_4 - T_1)}{C_{p,av}(T_3 - T_2)}\right) \tag{7.46}$$

Upon rearrangement, Eq. 7.46 reduces to

$$\eta_{\text{Brayton}} = 1 - \left(\frac{T_1}{T_2}\right)\left(\frac{T_4/T_1 - 1}{T_3/T_2 - 1}\right) \tag{7.47}$$

Figure 7.7 shows that processes 1–2 and 3–4 are isentropic, and $P_2 = P_3$ and $P_4 = P_1$, thus the previously derived equations for isentropic process are expressed by

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} \text{and} \left(\frac{T_3}{T_4}\right) = \left(\frac{P_3}{P_4}\right)^{(\gamma-1)/\gamma}$$

Therefore, the thermal efficiency is estimated by

$$\eta_{\text{Brayton}} = 1 - \left(\frac{1}{r_p^{(\gamma-1)/\gamma}} \right) \quad \text{(for constant heat capacity)} \tag{7.48}$$

where r_p is the compression ratio (P_2/P_1) and $\gamma = C_p/C_v$. Equation 7.48 shows that the Brayton cycle depends on the pressure ratio of the gas-turbine and the ratio of specific heats of the working fluid. The typical values of pressure ratio change between 5 and 20 [5]. As seen from Fig. 7.7, part of the electricity produced is used to drive the compressor. The back work ratio r_{bw} shows the part of the produced energy is diverted to the compressor

$$r_{\text{bw}} = \frac{W_{\text{comp.in}}}{W_{\text{turb.out}}} \tag{7.49}$$

Example 7.13 illustrates the analysis of an ideal Brayton cycle when the heat capacity is temperature dependent, while Example 7.14 illustrates the analysis of a real Brayton cycle. Example 7.15 illustrates the analysis of an ideal Brayton cycle with constant specific heats.

Example 7.13 Simple ideal Brayton cycle calculations with variable specific heats
A power plant is operating on an ideal Brayton cycle with a pressure ratio of $r_p = 9$. The fresh air temperature is 295 K at the compressor inlet and 1,300 K at the end of the compressor (inlet of the turbine). Using the standard-air assumptions, determine the thermal efficiency of the cycle.

Solution:
Assume that the cycle is at steady-state flow and the changes in kinetic and potential energy are negligible. Heat capacity of air is temperature dependent, and the air is an ideal-gas.
Consider Fig. 7.7.
Basis: 1 kg air. Using the data from the Appendix: Table D1
Process 1–2 isentropic compression
At T_1 = 295 K, H_1 = 295.17 kJ/kg; (P_r is the relative pressure defined in Eq. 7.36).

$$\frac{P_{r2}}{P_{r1}} = \frac{P_2}{P_1} = P_r \rightarrow P_{r2} = (9)(1.3068) = 11.76$$

Approximate values from Table D1 for the compressor exit at P_{r2} = 11.76
T_2 = 550 K and H_2 = 555.74 kJ/kg
Process 3–4 isentropic expansion in the turbine as seen on the *TS* diagram in Fig. 7.7.
T_3 = 1,300 K, H_3 = 1,395.97 kJ/kg; P_{r3} = 330.9 (From Table D1)

$$\frac{P_{r4}}{P_{r3}} = \frac{P_4}{P_3} \rightarrow P_{r4} = \left(\frac{1}{9}\right)(330.9) = 36.76$$

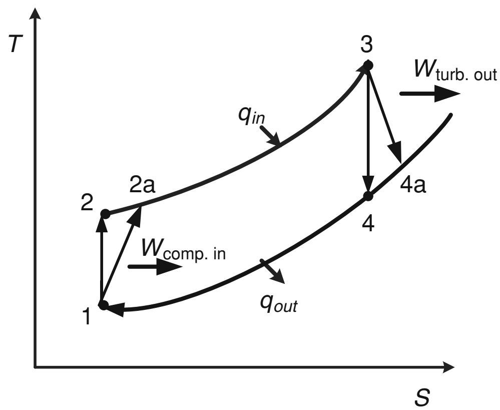
Approximate values from Table D1 at the exit of turbine P_{r4} = 36.76:
T_4 = 745 K and H_4 = 761.87 kJ/kg
The work input to the compressor:
$W_{\text{comp.in}} = H_2 - H_1 = (555.74 - 295.17)\text{ kJ/kg} = 260.6\text{ kJ/kg}$
The work output of the turbine:
$W_{\text{turb.out}} = H_4 - H_3 = (761.87 - 1{,}395.97)\text{ kJ/kg} = -634.1\text{ kJ/kg}$
The net work out: $W_{\text{net}} = W_{\text{out}} - W_{\text{in}} = -(634.1 - 260.6) = -373.53\text{ kJ/kg}$
The back work ratio r_{bw} becomes: $r_{\text{bw}} = \dfrac{W_{\text{comp.in}}}{W_{\text{turb.out}}} = \dfrac{260.6}{634.1} = 0.41$ or 41%
This shows that 41% of the turbine output has been used in the compressor.
The amount of heat added: $q_{\text{in}} = H_3 - H_2 = 1{,}395.97 - 555.74 = 840.23\text{ kJ/kg}$
The amount of heat rejected: $q_{\text{out}} = H_1 - H_4 = 295.17 - 761.87 = -466.7\text{ kJ/kg}$
The thermal efficiency: $\eta_{\text{th}} = \dfrac{W_{\text{net}}}{q_{\text{in}}} = 1 - \dfrac{q_{\text{out}}}{q_{\text{in}}} = \mathbf{0.444}$ **or (44.4%)**

Example 7.14 Thermal efficiency of an actual Brayton cycle with variable specific heats

A power plant is operating on an ideal Brayton cycle with a pressure ratio of $r_p = 9$. The fresh air temperature is 295 K at the compressor inlet and 1,300 K at the end of the compressor and at the inlet of the turbine. Assume the gas-turbine cycle operates with a compressor efficiency of 85% and a turbine efficiency of 85%. Determine the thermal efficiency of the cycle.

Solution:

Assume that the cycle is at steady-state flow and the changes in kinetic and potential energy are negligible. Heat capacity of air is temperature dependent, and the air is an ideal-gas. The standard-air assumptions are applicable
Basis: 1 kg air.



At T_1 = 295 K, H_1 = 295.17 kJ/kg; P_{r1} = 1.3068; (Table D1) P_r is the relative pressure (Eq. 7.36).

$$\frac{P_{r2}}{P_{r1}} = \frac{P_2}{P_1} = P_r \rightarrow P_{r2} = (9)(1.3068) = 11.76$$

Approximate values from Table D1 for the compressor exit at P_{r2} = 11.76:
T_2 = 550 K and H_2 = 555.74 kJ/kg
Process 3–4 isentropic expansion in the turbine as seen on the *TS* diagram above
T_3 = 1,300 K, H_3 = 1,395.97 kJ/kg; P_{r3} = 330.9 (From Table D1)

$$\frac{P_{r4}}{P_{r3}} = \frac{P_4}{P_3} \rightarrow P_{r4} = \left(\frac{1}{9}\right)(330.9) = 36.76$$

Approximate values from Table D1 at the exit of turbine at 36.76:
T_4 = 745 K and H_4 = 761.87 kJ/kg
The work input to the compressor:
$W_{\text{comp.in}} = H_2 - H_1 = (555.74 - 295.17)\,\text{kJ/kg} = 260.57\ \text{kJ/kg}$
The work output of the turbine:
$W_{\text{turb.out}} = H_4 - H_3 = (761.87 - 1,395.97)\,\text{kJ/kg} = -634.10\ \text{kJ/kg}$
From the efficiency definitions for compressor and turbine, we have

$$\eta_{\text{C}} = \frac{W_{Cs}}{W_{Ca}} \rightarrow W_{Ca} = \frac{W_{Cs}}{\eta_C} = \frac{260.6\ \text{kJ/kg}}{0.85}$$
$$= 306.6\ \text{kJ/kg (Actual compression work)}$$

(where $W_{\text{comp.in}} = W_{Cs} = 260.6$ kJ/kg in Example 7.13)
The work output of the turbine with a 85% efficiency

$$\eta_T = \frac{W_{Ta}}{W_{Ts}} \rightarrow W_{Ta} = \eta_T(W_{Ts}) = 0.85(634.1\ \text{kJ/kg}) = 539.0\text{kJ/kg}$$

(where$W_{\text{turb.out}} = W_s = 634.1$ kJ/kg in ideal operation in Example 7.13)
The net work out: $W_{\text{net}} = W_{\text{out}} - W_{\text{in}} = -(539.0 - 306.6) = -232.4$ kJ/kg
The back work ratio r_{bw} becomes: $r_{\text{bw}} = \dfrac{W_{\text{comp.in}}}{W_{\text{turb.out}}} = \dfrac{306.6}{539.0} = 0.568$
This shows that the compressor is now consuming 56.8% of the turbine output. The value of back work ratio increased from 41 to 56.8% because of friction, heat losses, and other non-ideal conditions in the cycle.
Enthalpy at the exit of compressor:

$$W_{Ca} = H_{2a} - H_1 \rightarrow H_{2\text{a}} = W_{Ca} + H_1 = (306.6 + 295.2)\text{kJ/kg} = 601.8\ \text{kJ/kg}$$

Heat added: $q_{\text{in}} = H_3 - H_{2a} = 1{,}395.97 - 601.8 = 794.2$ kJ/kg
The thermal efficiency: $\eta_{\text{th}} = \dfrac{W_{\text{net}}}{q_{\text{in}}} = \dfrac{232.4\text{kJ/kg}}{794.2\ \text{kJ/kg}} = \mathbf{0.292\ (or\ 29.2\%)}$
The actual Brayton-gas cycle thermal efficiency drops to 0.292 from 0.444. Efficiencies of the compressor and turbine affects the performance of the cycle. Therefore, for a better cycle thermal efficiency, significant improvements are necessary for the compressor and turbine operations.

	Ideal	Actual
W_{net}, kJ/kg	373.5	232.4
W_C, kJ/kg	260.6	306.6
W_T, kJ/kg	634.1	539.0
q_{in}, kJ/kg	840.2	794.2
q_{out}, kJ/kg	466.7	561.8
η_{th} %	44.4	29.2

Example 7.15 Ideal Brayton cycle with constant specific heats
A power plant is operating on an ideal Brayton cycle with a pressure ratio of $r_p = 9$. The air temperature is 300 K at the compressor inlet and 1,200 K at the end of the compressor. Using the standard-air assumptions and $\gamma = 1.4$ determine the thermal efficiency of the cycle.

Solution:
Assume that the cycle is at steady-state flow and the changes in kinetic and potential energy are negligible. The specific heat capacities are constant, and the air is an ideal-gas.

Consider Fig. 7.7.
Basis: 1 kg air and $\gamma = 1.4$.
Using the data from the Appendix: Table D1
Process 1–2 isentropic compression
$T_1 = 300$ K, $H_1 = 300.2$ kJ/kg;
From Eq. 7.34: $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} \rightarrow T_2 = 300(9)^{0.4/1.4} = 562\ \mathrm{K}$
Process 3–4 isentropic expansion in the turbine as seen on the *TS* diagram above

$$T_3 = 1{,}200\ \mathrm{K\ and}\ \frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{(\gamma-1)/\gamma} \rightarrow T_4 = 1{,}200\left(\frac{1}{9}\right)^{0.4/1.4} \rightarrow T_4 = 640.5\ \mathrm{K}$$

From Eq. 7.46: $\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)_4}$

$$= 1 - \frac{640.5 - 300}{1{,}200 - 562} = \mathbf{0.466\ or\ 46.6\%}$$

From Eq. 7.48: $\eta_{Brayton} = 1 - \left(\frac{1}{r_p^{(\gamma-1)/\gamma}}\right) = \mathbf{0.463\ or\ 46.3\%}$

The results of efficiency calculations are close to each other; for a constant heat capacity it is easy to use Eq. 7.48 directly to estimate the efficiency using the compression ratio r_p.

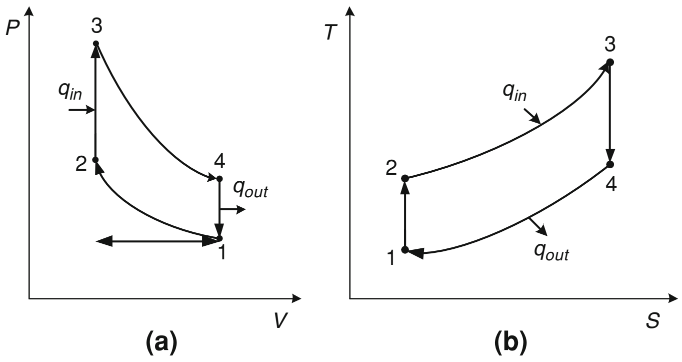


Fig. 7.9 Thermodynamic diagrams of Otto cycle; **a** Otto cycle on a *PV* diagram, **b** Otto cycle on a *TS* diagram

7.8.8 Otto Engine Efficiency

An *Otto cycle* is named after Nikolaus A. Otto who manufactured a four-stroke engine in 1876 in Germany. Otto cycle describes the functioning of a typical ideal cycle for spark-ignition reciprocating piston engine. This cycle is common in automobile internal combustion engines using a chemical fuel with oxygen from the air. The combustion process results heat, steam, carbon dioxide, and other chemicals at very high temperature. Petroleum fractions, such as diesel fuel, gasoline and petroleum gases are the most common fuels. Bioethanol, biodiesel, and hydrogen can also be used as fuels with modified engines.

Internal combustion engines require either spark ignition or compression ignition of the compressed air–fuel mixture [5, 15]. Gasoline engines take in a mixture of air and gasoline and compress it to around 12.8 bar (1.28 MPa), then use a high-voltage electric spark to ignite the mixture. The compression level in diesel engines is usually twice or more than a gasoline engine. Diesel engines will take in air only, and shortly before peak compression, a small quantity of diesel fuel is sprayed into the cylinder via a fuel injector that allows the fuel to instantly ignite. Most diesels also have a battery and charging system. Diesel engines are generally heavier, noisier, and more powerful than gasoline engines. They are also more fuel-efficient in most circumstances and are used in heavy road vehicles, some automobiles, ships, railway locomotives, and light aircraft.

Figure 7.9 shows the Otto engine cycles on *PV* and *TS* diagrams. The ideal cycle processes are:

- Process 1–2 is an isentropic compression of the air as the piston moves from bottom dead center to top dead center (*Intake stroke*).
- Process 2–3 is a constant-volume heat transfer to the air from an external source while the piston is at top dead center. This process leads to the ignition of the fuel–air mixture (*Compression stroke*).
- Process 3–4 is an isentropic expansion (*Power stroke*).

- Process 4–1 completes the cycle by a constant-volume process in which heat is rejected from the air while the piston is at bottom dead center (*Exhaust stroke*).

In the case of a four-stroke Otto cycle, technically there are two additional processes as shown in Fig. 7.9a: one for the exhaust of waste heat and combustion products (by isobaric compression), and one for the intake of cool oxygen-rich air (by isobaric expansion). However, these are often omitted in a simplified analysis. Processes 1–2 and 3–4 do work on the system but no heat transfer occurs during adiabatic expansion and compression. Processes 2–3 and 4–1 are isochoric (constant-volume) therefore heat transfer occurs but no work is done as the piston volume does not change.

Idealized *PV* diagram in Fig. 7.9b of the Otto cycle shows the combustion heat input q_{in} and the waste exhaust output q_{out}. The power stroke is the top curved line and the bottom is the compression stroke.

The Otto cycle is analyzed as a close system. The energy balance on a unit-mass basis is

$$\Delta U = (q_{in} - q_{out}) + (W_{in} - W_{out}) \tag{7.50}$$

Assuming that the heat capacity is constant, thermal efficiency can be derived by the heat added q_{in} and heat rejected q_{out}

$$\eta_{Otto} = 1 - \left(\frac{q_{out}}{q_{in}}\right) = 1 - \left(\frac{C_{v,av}(T_4 - T_1)}{C_{v,av}(T_3 - T_2)}\right) \tag{7.51}$$

In an ideal Otto cycle, there is no heat transfer during the process 1–2 and 3–4 as they are reversible adiabatic processes. Heat is supplied only during the constant-volume processes 2–3 and heat is rejected only during the constant-volume processes 4–1.

Upon rearrangement:

$$\eta_{Otto} = 1 - \left(\frac{T_1}{T_2}\right)\left(\frac{T_4/T_1 - 1}{T_3/T_2 - 1}\right) \tag{7.52}$$

Since $T_4/T_1 = T_3/T_2$, (see Fig. 7.8), Eq. (7.52) reduces to:

$$\eta_{Otto} = 1 - \left(\frac{T_1}{T_2}\right) \tag{7.53}$$

From the isentropic equations of ideal gases, we have

$$C_v \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{V_2}{V_1}\right) = 0 \tag{7.54}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = r^{(\gamma-1)} \tag{7.55}$$

where r is the compression ratio (V_1/V_2) and $\gamma = C_p/C_v$, and $R = C_p–C_v$ for an ideal-gas. Then the thermal efficiency can be expressed as

$$\eta_{\text{Otto}} = 1 - \left(\frac{1}{r^{(\gamma-1)}}\right) \quad \text{(constant specific heats)} \tag{7.56}$$

The specific heat ratio of the air–fuel mixture γ varies somewhat with the fuel, but is generally close to the air value of 1.4, and when this approximation is used the cycle is called an *air-standard cycle*. However, the real value of γ for the combustion products of the fuel/air mixture is approximately 1.3.

Equation 7.56 shows that the Otto cycle depends upon the compression ratio $r = V_1/V_2$. At the higher compression ratio, the efficiency is higher and the temperature in the cylinder is higher. However the maximum compression ratio is limited approximately to 10:1 for typical automobiles. Usually this does not increase much because of the possibility of auto ignition, which occurs when the temperature of the compressed fuel/air mixture becomes too high before it is ignited by the flame front leading to *engine knocking*. Example 7.16 illustrates the analysis of an ideal Otto engine with temperature-dependent heat capacities, while Example 7.17 illustrates with constant heat capacities.

Example 7.16 Efficiency calculations of ideal Otto engine with variable specific heats

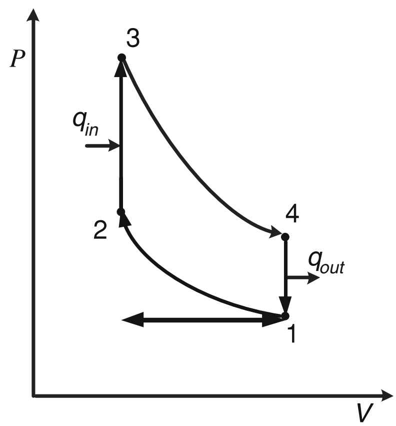
An ideal Otto cycle operates with a compression ratio ($r = V_{max}/V_{min}$) of 8.8. Air is at 101.3 kPa and 280 K at the start of compression (state 1). During the constant-volume heat-addition process, 1,000 kJ/kg of heat is transferred into the air from a source at 1,900 K. Heat is discharged to the surroundings at 280 K. Determine the thermal efficiency of energy conversion.

Solution:

Assume that the surroundings are at 280 K and the kinetic and potential energy changes are negligible. The specific heats are temperature dependent.

Heat capacity of air is temperature dependent, and the air is an ideal-gas. The *PV* diagram below shows the cycle and the four processes

Processes in the cycle:



1–2 Isentropic compression
2–3 Constant-volume heat transfer
3–4 Isentropic expansion
4–1 Constant-volume heat discharge

Basis: 1 kg air. Using the data from the Appendix:
q_{in} = 1,000 kJ/kg
U_1 = 199.75 kJ/kg; V_{r1} = 783.0 at T = 280 K From Table D1
(V_r is the relative specific volume defined in Eq. 7.38)

$$\frac{V_{r2}}{V_{r1}} = \frac{V_2}{V_1} = \frac{1}{8.8} \rightarrow V_{r2} = \frac{V_{r1}}{r} = \frac{783}{8.8} = 88.97$$

At the value of V_{r2} = 88.97, the air properties from Table D1: U_2 = 465.5 kJ/kg and T_2 = 640 K

From isentropic compression of air, we estimate

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)\left(\frac{V_1}{V_2}\right) = 101.3\left(\frac{640}{280}\right)8.8 \; = 2{,}037 \text{ kPa(ideal-gas equation)}$$

The heat transferred in the path 2–3:

$$q_{\text{in}} = 1{,}000 \text{ kJ/kg} = U_3 - U_2 \rightarrow U_3 \; = 1465.5 \text{ kJ/kg}$$

At U_3 = 1,465.5 kJ kg, we estimate T_3 and V_{r3} by interpolation using the data below

T (K)	U (kJ/kg)	V_r
1,750	1,439.8	4.328
1,800	1,487.2	3.994
1,777	1,465.5	4.147

V_{r3} = 4.147 at T_3 = 1777 k

We estimate the pressure at state 3 as before

$$P_3 = P_2\left(\frac{T_3}{T_2}\right)\left(\frac{V_2}{V_3}\right) = 2{,}037\left(\frac{1{,}777}{640}\right)(1) \; = 5{,}656 \text{ kPa}$$

Internal energy of air at state 4.

$$\left(\frac{V_{r4}}{V_{r3}}\right) = \left(\frac{V_4}{V_3}\right) = 8.8 \rightarrow V_{r4} = (8.8)(4.147) \; = 36.5$$

At V_{r4} = 36.5, approximate values from Table D1 for: U_4 = 658 kJ/kg and T_4 = 880 K
The process 4–1 is a constant heat discharge. We estimate the discharged heat q_{out}

$$q_{\text{out}} = U_1 - U_4 \rightarrow q_{\text{out}} \; = -458.2 \text{ kJ/kg}$$

$$P_4 = P_3\left(\frac{T_4}{T_3}\right)\left(\frac{V_3}{V_4}\right) = 5{,}656\left(\frac{880}{1{,}777}\right)\left(\frac{1}{8.8}\right) = 318.3 \text{ kPa}$$

The net heat transfer is equal to the net work output:

$$W_{\text{net}} = q_{\text{in}} - q_{out} = (1{,}000 - 458.2)\,\text{kJ/kg} = 541.8\,\text{kJ/kg}$$

So the thermal efficiency: $\eta_{\text{th}} = \frac{W_{\text{net}}}{q_{\text{in}}} = \mathbf{0.542 \text{ or } 54.2\%}$

Although all the processes are internally reversible, the heat transfer and discharge take place at finite temperature difference, and are irreversible.

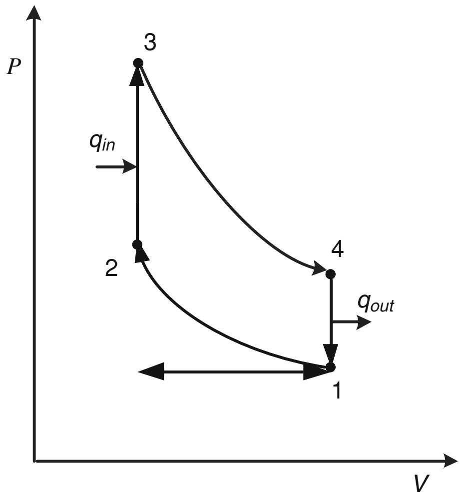
Example 7.17 Efficiency calculations of an ideal Otto cycle with constant specific heats

An ideal Otto cycle operates with a compression ratio ($V_{\text{max}}/V_{\text{min}}$) of 8. Air is at 101.3 kPa and 300 K at the start of compression (state 1). During the constant-volume heat-addition process, 730 kJ/kg of heat is transferred into the air from a source at 1,900 K. Heat is discharged to the surroundings at 300 K. Determine the thermal efficiency of energy conversion. The average specific heats: $C_{p,av}$ = 1.00 kJ/kg K and $C_{v,av}$ = 0.717 kJ/kg K.

Solution:

Assume: the air-standard assumptions are applicable. The changes in kinetic and potential energy s are negligible. The specific heats are constant.

The *PV* diagram below shows the cycle and the four states



Processes in the cycle:

1–2 Isentropic compression

2–3 Constant-volume heat transfer

3–4 Isentropic expansion

4–1 Constant-volume heat discharge

Basis: 1 kg air.
q_{in} = 730 kJ/kg

Using the data from Table D1 at T_o = 300 K: U_1 = 214.1 kJ/kg.
The average specific heats:
$C_{p,av}$ = 1.0 kJ/kg K and $C_{p,av}$ = 0.717 kJ/kg K and γ = 1/0.717 = 1.4.
From Eq. 7.34:

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \rightarrow T_2 = \frac{V_1}{V_2} T_1 = 8(300 \text{ K})^{0.4} = 689 \text{ K}$$

Estimate T_3 from the heat transferred at constant-volume in process 2–3:

$$q_{in} = U_3 - U_2 = C_{v,av}(T_3 - T_2) \rightarrow T_3 = 1{,}703 \text{ K}$$

Process 3–4 isentropic expansion: $T_4 = T_3\left(\frac{V_3}{V_4}\right)^{\gamma-1} = 1{,}703.0\left(\frac{1}{8}\right)^{0.4} = 741.3$ K
Process 4–1: $q_{out} = U_4 - U_1 = C_{v,av}(T_4 - T_1) = 316.4 \text{ kJ/kg}$, q_{in} = 730 kJ/kg
Net heat transfer is equal to the net work output:
$W_{net} = q_{in} - q_{out} = 730.0 - 316.4 = 413.6\,\text{kJ/kg}$
From Eq. 7.56: So the thermal efficiency is
$$\eta_{Otto} = 1 - \left(\frac{1}{r^{(\gamma-1)}}\right) = 0.564 \text{ (or } 56.4\%)$$

7.8.9 Diesel Engine Efficiency

Figure 7.10 shows an ideal Diesel cycle on the *PV* and *TS* diagrams. The compression-ignited engine is first proposed by Rudolph Diesel in 1890. Most truck and automotive diesel engines use a similar cycle to spark-ignited gasoline engine, but with a compression heating ignition system, rather than needing a separate ignition system. In combustion-ignited engine, the air is compressed to a temperature that is above the autoignition temperature of the fuel. Therefore the combustion starts as the fuel is injected into the hot air at constant pressure. This variation is called the diesel cycle.

The Diesel cycle is analyzed as a piston-cylinder of a close system. The amount of heat transferred to the working fluid air at constant pressure on a unit-mass basis is

$$q_{in} = P_2(V_3 - V_2) + (U_3 - U_2) = H_3 - H_2 = C_{p,av}(T_3 - T_2) \tag{7.57}$$

The amount of heat rejected at constant-volume is

$$q_{out} = U_1 - U_4 = C_{v,av}(T_1 - T_4) \tag{7.58}$$

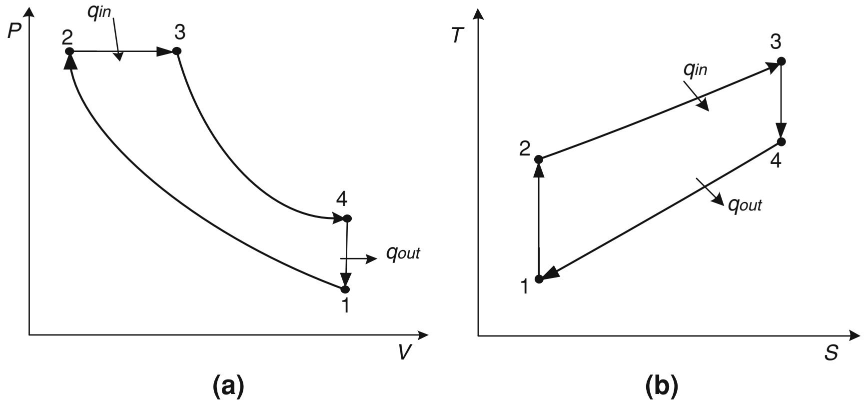


Fig. 7.10 Ideal Diesel cycle **a** on *PV* diagram, **b** on *TS* diagram; the cycle follows the numbers 1–4 in clockwise direction

Alternatively, thermal efficiency can be derived by the heat added q_{in} and heat rejected q_{out}

$$\eta_{\text{Diesel}} = 1 - \left(\frac{q_{\text{out}}}{q_{\text{in}}}\right) = 1 - \left(\frac{(T_4 - T_1)}{\gamma(T_3 - T_2)}\right) = 1 - \left(\frac{T_1(T_4/T_1 - 1)}{\gamma T_2(T_3/T_2 - 1)}\right) \tag{7.59}$$

where $\gamma = C_p/C_v$. In an ideal Diesel cycle, the cutoff ratio r_c is defined as the ratio of the volume after and before the combustion

$$r_c = \frac{V_3}{V_2} \tag{7.60}$$

After using the cutoff ratio r_c and compression ratio $r = V_1/V_2$ with the isentropic ideal-gas relations, given in Eq. 7.33, the thermal efficiency of energy conversion becomes

$$\eta_{\text{Diesel}} = 1 - \frac{1}{r^{(\gamma-1)}}\left(\frac{r_c^{\gamma} - 1}{\gamma(r_c - 1)}\right) \text{ (With constant specific heats)} \tag{7.61}$$

Equation 7.61 shows that as the cutoff ratio r_c decreases, the efficiency of the Diesel cycle increases. For the limiting ratio of $r_c = 1$, the value of the bracket in Eq. 7.61 becomes unity, and the efficiencies of the Otto and Diesel cycles become identical. Thermal efficiencies of large diesel engines vary from 35 to 40% [5, 15]. Example 7.18 illustrates the analysis of diesel engine with constant heat capacity, while Example 7.19 with temperature dependent heat capacity.

Example 7.18 Thermal efficiency of an ideal Diesel engine with the constant specific heats

An ideal Diesel cycle has an air-compression ratio of 20 and a cutoff ratio of 2. At the beginning of the compression, the fluid pressure, temperature, and volume

are 14.7 psia, 70°F, and 120 in^3, respectively. The average specific heats of air at room temperature are $C_{p,av}$ = 0.24 Btu/lb R and, $C_{v,av}$ = 0.171 Btu/lb R. Estimate the thermal efficiency with the cold-air-standard assumptions.

Solution:
Assume: the cold-air-standard assumptions are applicable. Air is ideal-gas. The changes in kinetic and potential energies are negligible.

Consider Fig. 7.10.

P_1 = 14.7 psia, T_1 = 70°F, and V_1 = 120 in^3, $C_{p,av}$ = 0.24 Btu/lb R and, $C_{v,av}$ = 0.171 Btu/lb R, R = 10.73 psia ft^3/lbmol R, MW = 29 lb/lbmol

$\gamma = C_{p,av}/C_{v,av}$ = 1.4, r = 20 and r_c = 2.
The air mass: $m_{\text{air}} = MW\dfrac{PV}{RT} = 0.0052\,\text{lb}$
The volumes for each process: $V_1 = V_4$ =120 in^3
$V_2 = V_1/r$ = 6 in^3
$V_3 = V_2\ r_c$ = 12 in^3
Process 1–2: isentropic compression: $\left(\dfrac{T_2}{T_1}\right) = \left(\dfrac{V_1}{V_2}\right)^{(\gamma-1)} \rightarrow T_2 = 1{,}756.6$ R
(Eq. 7.34)
Process 2–3: Heat addition at constant pressure for ideal gas:
$\left(\dfrac{T_3}{T_2}\right) = \left(\dfrac{V_3}{V_2}\right) \rightarrow T_3 = 3{,}513$ R
$q_{\text{in}} = m(H_3 - H_2) = mC_{p,av}(T_3 - T_2)\ = 2.19\,\text{Btu}$
Process 3–4: isentropic expansion: $\left(\frac{T_4}{T_3}\right) = \left(\frac{V_3}{V_4}\right)^{(\gamma-1)} \rightarrow T_4 = 1,398$ R
Process 4–1: heat rejection at constant-volume for ideal-gas
$q_{\text{out}} = m(H_1 - H_4) = mC_{v,av}(T_4 - T_1)\ = 0.772\,\text{Btu}$
The net work output is equal to the difference between heat input (q_{in}) and waste heat (q_{out})
$W_{\text{out}} = q_{\text{in}} - q_{\text{out}}\ = (2.19 - 0.772) = 1.418$ Btu
Thermal efficiency: $\eta_{\text{th}} = \dfrac{W_{\text{net}}}{q_{\text{in}}} = \mathbf{0.647}$ (**or 64.7**%)
Thermal efficiency can also be estimated from Eq. (7.61):

$$\eta_{\text{Diesel}} = 1 - \frac{1}{r^{(\gamma-1)}}\left(\frac{r_c^{\gamma} - 1}{\gamma(r_c - 1)}\right) = \mathbf{0.647}(\textbf{or 64.7}\%)$$

Example 7.19 Thermal efficiency of an ideal Diesel engine with variable specific heats
An ideal Diesel cycle has an air-compression ratio of 18 and a cutoff ratio of 2. At the beginning of the compression, the fluid pressure and temperature are 100 kPa,

300 K, respectively. Utilizing the cold-air-standard assumptions, determine the thermal efficiency.

Solution:
Assume: the cold-air-standard assumptions are applicable. Air is ideal-gas. The changes in kinetic and potential energies are negligible. The specific heats depend on temperature.
Consider Fig 7.10.
P_1 = 100 kPa, T_1 = 300 K, $r = V_1/V_2 = 18$ and $r_c = V_3/V_2 = 2$.
At 300 K, U_1 = 214.1 kJ/kg, V_{r1} = 621.2
(V_r is the relative specific volume) (Table D1)

From Eq. 7.38:

$$\frac{V_{r2}}{V_{r1}} = \frac{V_2}{V_1} = \frac{1}{18} \rightarrow V_{r2} = \frac{V_{r1}}{r} = \frac{621.2}{18} = 34.5$$

At this value of V_{r2}= 34.5, approximate values for:
H_2 = 932.9 kJ/(kg K), T_2 = 900 K (Table D1)

The heat transferred at constant pressure in process 2–3:

$$\frac{V_3}{T_3} = \frac{V_2}{T_2} = \longrightarrow T_3 = \frac{V_3}{V_2} T_2 = 2T_2 = 1800 \text{ K}$$

$$H_3 = 2003.3 \text{ kJ/kg}, \; V_{r3} = 4.0 \text{ (Table D1)}$$

$$q_{\text{in}} = H_3 - H_2 = 1{,}070.4 \text{ kJ/kg}$$

Processes 3–4: Next, we need to estimate the internal energy of air at state 4:

$$\left(\frac{V_{r4}}{V_{r3}}\right) = \left(\frac{V_4}{V_3}\right) = \left(\frac{V_4}{2V_2}\right) = \left(\frac{18}{2}\right) \rightarrow V_{r4} = (9)(4.0) = 36$$

At V_{r4} = 36, U_4 = 659.7 kJ/kg (Table D1)

The process 4–1 is heat discharge at constant-volume. We estimate the discharged heat q_{out}

$$q_{\text{out}} = U_4 - U_1 = 659.7 - 214.1 = 445.6 \text{ kJ/kg}$$

The net heat transfer is equal to the net work output:

$$W_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1{,}070.4 - 445.6 = 624.8 \text{ kJ/kg}$$

So the thermal efficiency is $\eta_{\text{th}} = \dfrac{W_{\text{net}}}{q_{\text{in}}} = \mathbf{0.583 \text{ or } 58.3\%}$

7.8.10 Ericsson and Stirling Engine Efficiency

The Ericsson engine is an "*external combustion engine*" because it is externally heated. The four processes that occur in an ideal Ericsson cycle are:

- Process 1–2: isothermal compression. The compressed air flows into a storage tank at constant pressure.
- Process 2–3: isobaric heat addition. The compressed air flows through the regenerator and picks up heat.
- Process 3–4: isothermal expansion. The cylinder expansion-space is heated externally, and the gas undergoes isothermal expansion.
- Process 4–1: isobaric heat removal. Before the air is released as exhaust, it is passed back through the regenerator, thus cooling the gas at a low constant pressure, and heating the regenerator for the next cycle.

A *Stirling engine* operates by cyclic compression and expansion of air or other gas at different temperature levels and converts the heat of hot gas into mechanical work. The Stirling engine has high efficiency compared to steam engines and can use almost any heat source [14, 15, 29]. Stirling cycle has four totally reversible processes:

- Process 1–2: isothermal heat addition from external source
- Process 2–3: internal heat transfer from working fluid to regenerator at constant-volume
- Process 3–4: isothermal heat rejection
- Process 4–1: internal heat transfer from regenerator back to working fluid at constant-volume.

Heat addition during process 1–2 at T_H and heat rejection process 3–4 at T_C are both isothermal. For a reversible isothermal process heat transfer is estimated by

$$q = T\Delta S \tag{7.62}$$

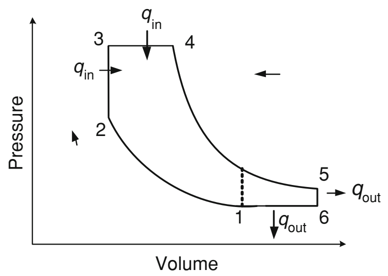
The change in entropy of an ideal-gas at isothermal conditions is

$$\Delta S_{1-2} = -R\ln\frac{P_2}{P_1} \tag{7.63}$$

Using Eqs. 7.62 and 7.63, the heat input and output are estimated by

$$q_{\text{in}} = T_H(S_2 - S_1) = T_H\left(-R\ln\frac{P_2}{P_1}\right) = RT_H\ln\frac{P_1}{P_2} \tag{7.64}$$

$$q_{out} = T_C(S_4 - S_3) = T_C\left(-R\ln\frac{P_4}{P_3}\right) = RT_C\ln\frac{P_3}{P_4} \tag{7.65}$$



The ideal Atkinson cycle consists of following operations:
1–2 Isentropic or reversible adiabatic compression
2–3 Isochoric heating
3–4 Isobaric heating
4–5 Isentropic expansion
5–6 Isochoric cooling
6–1 Isobaric cooling

Fig. 7.11 Ideal Atkinson gas cycle

The thermal efficiency of the Ericsson cycle becomes

$$\eta_{\text{Ericsson}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{RT_C \ln(P_4/P_3)}{RT_H \ln(P_1/P_2)} = 1 - \frac{T_C}{T_H} \tag{7.66}$$

Since $P_1 = P_4$ and $P_3 = P_2$.

Both the ideal Ericsson and Stirling cycles are external combustion engines. Both the cycles are totally reversible as is the Carnot cycle and have the same thermal efficiency between the same temperature limits when using an ideal-gas as the working fluid

$$\eta_{\text{Ericsson}} = \eta_{\text{Stirling}} = \eta_{\text{Carnot}} = 1 - \frac{T_C}{T_H} \tag{7.67}$$

Both the cycles utilize regeneration to improve efficiency. Between the compressor and the expander heat is transferred to a thermal energy storage device called the regenerator during one part of the cycle and is transferred back to the working fluid during another part of the cycle. Equation 7.67 is valid for both the closed and steady-state flow cycles of the engines.

7.8.11 Atkinson Engine Efficiency

The *Atkinson engine* is an internal combustion engine and is used in some modern hybrid electric applications. Figure 7.11 shows the cyclic processes in the Atkinson engine. In the engine, the expansion ratio can differ from the compression ratio and the engine can achieve greater thermal efficiency than a traditional piston engine [15]. Expansion ratios are obtained from the ratio of the combustion chamber volumes when the piston is at bottom dead center and top dead center

In Atkinson cycle, the intake valve allows a reverse flow of intake air. The goal of the modern Atkinson cycle is to allow the pressure in the combustion chamber at the end of the power stroke to be equal to atmospheric pressure; when this occurs, all the available energy has been utilized from the combustion process. For

any given portion of air, the greater expansion ratio allows more energy to be converted from heat to useful mechanical energy, hence the engine is more efficient. The disadvantage of the four-stroke Atkinson cycle is the reduced power output as a smaller portion of the compression stroke is used to compress the intake air. Atkinson cycles with a supercharger to make up for the loss of power density are known as *Miller engines* [15, 29]. The power of the Atkinson engine can be supplemented by an electric motor. This forms the basis of an Atkinson cycle-based hybrid electric drivetrain.

7.9 Improving Efficiency of Heat Engines

The followings may be some options toward increasing the efficiency of heat engines [4, 10, 13, 21]:

- Increased hot side temperature is the approach used in modern combined-cycle gas turbines. The melting point of the construction of materials and environmental concerns regarding NO_x production limit the maximum temperature on the heat engines.
- Lowering the output temperature by using mixed chemical working fluids, such as using a 70/30 mix of ammonia and water as its working fluid may also increase the efficiency. This mixture allows the cycle to generate useful power at considerably lower temperatures.
- Use of supercritical fluids, such as CO_2 as working fluids may increase the efficiency.
- Exploit the chemical properties of the working fluid, such as nitrogen dioxide (NO_2), which has a natural dimer as di-nitrogen tetra oxide (N_2O_4). At low temperature, the N_2O_4 is compressed and then heated. The increasing temperature causes each N_2O_4 to break apart into two NO_2 molecules. This lowers the molecular weight of the working fluid, which drastically increases the efficiency of the cycle. Once the NO_2 has expanded through the turbine, it is cooled and recombined into N_2O_4. This is then fed back to the compressor for another cycle.
- New fuels, such as hydrogen, may have positive impact on the performance of the engines since the energy density of hydrogen is considerably higher than that of electric batteries.

7.10 Hydroelectricity

Hydroelectricity is the production of electrical power through the use of the gravitational force of falling or flowing water. Figure 7.12 shows a schematic of a conventional hydroelectric dam, in which hydroelectric power comes from the

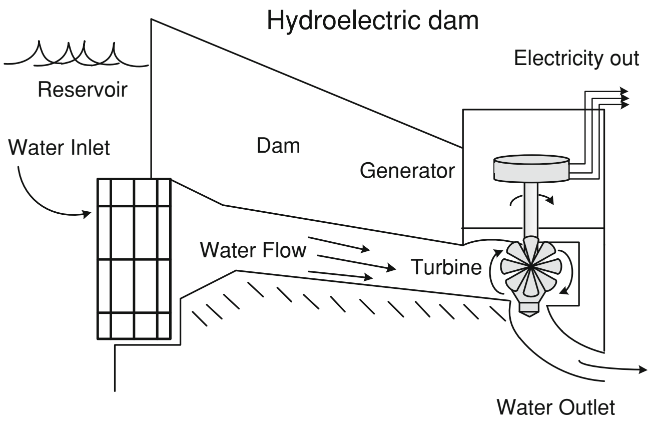


Fig. 7.12 Cross-section of a conventional hydroelectric dam

potential energy of dammed water driving a water turbine and generator. The power extracted from the water depends on the volume and on the difference in height between the source and the water's outflow. This height difference is called the head. The amount of potential energy in water is proportional to the head. A large pipe called the penstock delivers water to the turbine. Example 7.20 illustrates the efficiency calculations for a hydraulic turbine.

Pumped-storage method produces electricity to supply high peak demands by moving water between reservoirs at different elevations. At times of low electrical demand, excess generation capacity is used to pump water into the higher reservoir. When there is higher demand, water is released back into the lower reservoir through a turbine. Compared to wind farms, hydroelectricity power plants have a more predictable load factor. If the project has a storage reservoir, it can generate power when needed. Hydroelectric plants can be easily regulated to follow variations in power demand [12]. One method of meeting the additional electric power demand at peak usage is to pump some water from a source such as a lake back to the reservoir of a hydropower plant at a higher elevation when the demand or the cost of electricity is low. Example 7.21 illustrates the pumped energy in a hydropower plant.

Example 7.20 Efficiency of a hydraulic turbine
Electricity is produced by a hydraulic turbine installed near a large lake. Average depth of the water in the lake is 45 m. The water mass flow rate is 600 kg/s. The produced electric power is 220 kW. The generator efficiency is 95%. Determine the overall mechanical efficiency of the turbine-generator and the shaft work transferred from the turbine to the generator.

Solution:
Assume: the mechanical energy of water at the turbine exit is small and negligible. The density of the water is 1,000 kg/m^3.

$\dot{m} = 600\text{kg/s}, \eta_{gen} = 0.95$

$\dot{W}_{out} = 220\,\text{kW}$

$$\dot{W}_{fluid} = \dot{m}gz = (600\,\text{kg/s})\left(9.81\,\text{m/s}^2\right)(45\,\text{m})\left(\text{kJ/kg} = 1000\,\text{m}^2/\text{s}^2\right) = 264.9\ \text{kW}$$

The overall mechanical efficiency of the turbine-generator:

$$\eta_{overall} = \eta_{turb}\eta_{gen} = \frac{\text{energy out}}{\text{energy in}} = \frac{220\,\text{kW}}{264.9\ \text{kW}} = \mathbf{0.83}$$

$\eta_{gen} = 0.95$

$$\eta_{overall} = \eta_{turb}\eta_{gen} = 0.83 \rightarrow \ \eta_{turb} = \frac{\eta_{overall}}{\eta_{gen}} = 0.87$$

The shaft power transferred from the turbine:

$$\dot{W}_{turb} = \eta_{turb}\dot{W}_{fluid} = 0.87\ (264.9\,\text{kW}) = 230.5\,\text{kW}$$

The lake supplies 264.9 kW of mechanical energy to the turbine. Only 87% of the supplied energy is converted to shaft work. This shaft work drives the generator and 220 kW is produced.

Example 7.21 Pumped energy in a hydropower plant
Consider a hydropower plant reservoir with an energy storage capacity of 1×10^6 kWh. This energy is to be stored at an average elevation of 60 m relative to the ground level. Estimate the minimum amount of water has to be pumped back to the reservoir.

Solution:

Assume that the evaporation of the water is negligible.

$PE = 1\times10^6$ kWh, $\Delta z = 60$ m, g = 9.8 m/s^2

Energy of the work potential of the water: $PE = mg\Delta z$
Amount of water:

$$m = \frac{PE}{g\Delta z} = \frac{1\times10^6\,\text{kWh}}{9.8\,\text{m/s}^2(60\,\text{m})}\left(\frac{3600\,\text{s}}{\text{h}}\right)\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right) = \mathbf{6.122\times10^9\,kg}$$

7.11 Wind Electricity

The power produced by a wind turbine is proportional to the kinetic energy of the wind captured by the wind turbine, and estimated by

$$\dot{W}_{\text{wind}} = \eta_{\text{wind}} \rho \frac{\pi v^3 D^2}{8} \tag{7.68}$$

where ρ is the density of air, v is the velocity of air, D is the diameter of the blades of the wind turbine, and η_{wind} is the efficiency of the wind turbine. Therefore, the power produced by the wind turbine is proportional to the cube of the wind velocity and the square of the blade span diameter. The strength of wind varies, and an average value for a given location does not alone indicate the amount of energy a wind turbine could produce there [24]. Example 7.22 illustrates the efficiency calculations for a wind turbine.

Example 7.22 Efficiency of a wind turbine

A wind turbine-generator with a 25-foot-diameter blade produces 0.5 kW of electric power. In the location of the wind turbine, the wind speed is 11 mile per hour. Determine the efficiency of the wind turbine-generator.

Solution:
Assume: the wind flow is steady. The wind flow is one-dimensional and incompressible. The frictional effects are negligible.
ρ_{air} = 0.076 lb/ft^3, D = 25 ft
Actual power production = 0.5 kW

Kinetic energy can be converted to work completely.
The power potential of the wind is its kinetic energy.

Average speed of the wind: $v_1 = (11\,\text{mph})\left(\frac{1.4667\,\text{ft/s}}{1\,\text{mph}}\right) = 16.13\,\text{ft/s}$

The mass flow rate of air: $\dot{m} = \rho v_1 A = \rho v_1 \frac{\pi D^2}{4} = 601.4\,\text{lb/s}$

$$\dot{W}_{\max} = \dot{m}\frac{v_1^2}{2g_c} = (601.4\,\text{lb/s})\frac{(16.13\,\text{ft/s})^2}{2}\left(\frac{1\,\text{lb}_\text{f}}{32.2\,\text{lb}_\text{m}\,\text{fts}^2}\right)\left(\frac{1\,\text{kW}}{737.56\,\text{lb}_\text{f}\,\text{ft/s}}\right) = 3.29\,\text{kW}$$

This is the available energy to the wind turbine. The turbine-generator efficiency is

$$\eta_{\text{wind turb}} = \frac{\dot{W}_{\text{act}}}{\dot{W}_{\max}} = \frac{0.5\,\text{kW}}{3.29\,\text{kW}} = \mathbf{0.152\ or\ 15.2\%}$$

Only 15.2% of the incoming kinetic energy is converted to electric power. The remaining part leaves the wind turbine as outgoing kinetic energy.

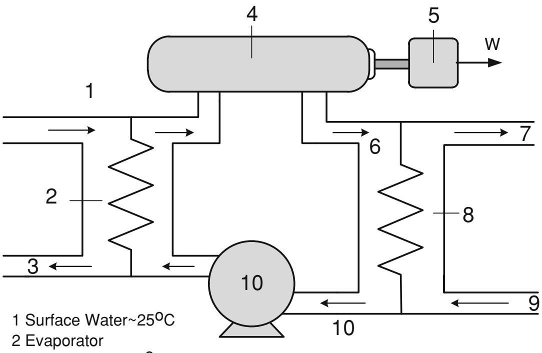


Fig. 7.13 Diagram of a closed-cycle ocean thermal energy conversion (OTEC) plant

7.12 Geothermal Electricity

Geothermal electricity refers to the energy conversion of geothermal energy to electric energy. Technologies in use include dry steam power plants, flash steam power plants, and binary cycle power plants. Geothermal power is considered to be sustainable because the heat extraction is small compared with the earth's heat content. Estimates of the geothermal electricity generating potential vary from 35 to 2,000 GW. Current worldwide installed capacity is around 11 GW.

Geothermal electric plants have until recently been built exclusively where high temperature geothermal resources are available near the surface. The development of binary cycle power plants and improvements in drilling and extraction technology may enable enhanced geothermal systems over a much greater geographical range [8].

The thermal efficiency of geothermal electric plants is low, around 10–23%, because geothermal fluids are at a low temperature compared with steam temperature from boilers. This low temperature limits the efficiency of heat engines in extracting useful energy during the generation of electricity. Exhaust heat is wasted, unless it can be used directly, for example, in greenhouses, timber mills, and district heating. In order to produce more energy than the pumps may consume, electricity generation requires high temperature geothermal fields and specialized heat cycles.

7.13 Ocean Thermal Energy Conversion

Ocean thermal energy conversion uses the difference between cooler deep and warmer shallow or surface ocean waters to run a heat engine and produce useful work, usually in the form of electricity (see Fig. 7.13). Warm surface seawater is

pumped through a heat exchanger to vaporize the fluid. The expanding vapor turns the turbo-generator. Cold water, pumped through a second heat exchanger, condenses the vapor into a liquid, which is then recycled through the system. In the tropics, the temperature difference between surface and deep water is a modest 20–25°C. Ocean thermal energy conversion systems is still considered an emerging technology with a thermal efficiency of 1–3%, which is well below the theoretical maximum for this temperature difference of between 6 and 7% [1, 17]. The most commonly used heat cycle for ocean thermal energy conversion systems is the Rankine cycle using a low-pressure turbine system. Closed-cycle engines use working fluids such as ammonia or R-134a. Open-cycle engines use vapor from the seawater itself as the working fluid.

7.14 Thermoelectric Effect

Thermoelectric effect involves energy conversions between heat and electricity as a temperature difference creates an electric potential or electric potential creates a heat flow leading to a temperature difference [31]. There are two thermoelectric effects:

- The *Seebeck effect* refers to conversion of heat to electricity.
- The *Peltier effect* refers to conversion of electricity to heat.

Measuring temperature by *thermocouples* operating between a hot and a cold junction is based on the Seebeck effect. A commonly used thermoelectric material in such applications is Bismuth telluride (Bi_2Te_3). The thermoelectric efficiency approaches to the Carnot limit.

Thermoelectric materials can be used as refrigerators, called "thermoelectric coolers", or "Peltier coolers", although it is far less common than vapor-compression refrigeration. Compared to a vapor-compression refrigerator, the main advantages of a Peltier cooler are its lack of moving parts or circulating liquid, and its small size and flexible shape. Another advantage is that Peltier coolers do not require refrigerant liquids, which can have harmful environmental effects. The main *disadvantage* of Peltier coolers is that they cannot simultaneously have low cost and high power efficiency. Advances in thermoelectric materials may lead to the creation of Peltier coolers that are both cheap and efficient.

7.15 Efficiency of Heat Pumps and Refrigerators

Heat pumps, refrigerators, and air conditioners use an outside work to move heat from a colder to a warmer region, so their function is the opposite of a heat engine. Since they are heat engines, these devices are also limited by the Carnot efficiency.

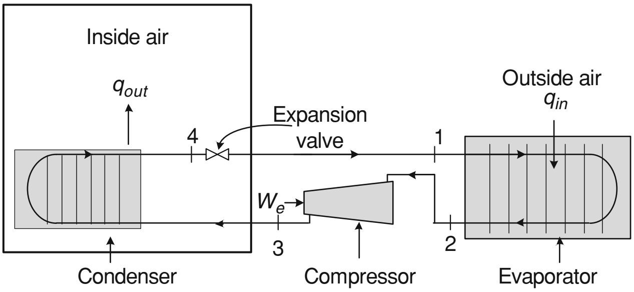


Fig. 7.14 A typical heat pump for heating a room; compressor pumps a gas from a low temperature region to a high temperature region. Within the evaporator, heat of vaporization is absorbed from the outside surroundings. Compressor increases the pressure and temperature of the vapor, and pumps it through the cycle. In the condenser, the vapor condenses and releases heat of condensation to the inside surroundings. The expansion valve causes a flash of the liquid to boil because of sudden drop in pressure; complete vaporization of the liquid takes place in the evaporator

A heat pump can be used for heating or cooling as part of *heating, ventilation, and air conditioning* [18] applications. The heat pump can heat and when necessary it uses the basic refrigeration cycle to cool. To do that a heat pump can change which coil is the condenser and which the evaporator by controlling the flow direction of the refrigerant by a reversing valve. So the heat can be pumped in either direction. In cooler climates it is common to have heat pumps that are designed only to heat. In heating mode, the outside heat exchanger is the evaporator and the indoor exchanger is the condenser to discharge heat to the inside air. In cooling mode, however, the outside heat exchanger becomes the condenser and the indoor exchanger is the evaporator to absorb heat from the inside air. The next sections discuss the heat pumps and refrigerators [5, 37]

7.15.1 Heat Pumps

A heat pump uses a fluid which absorbs heat as it vaporizes and releases the heat when it condenses. Figure 7.14 shows a typical heat pump drawing heat from the ambient air. A heat pump requires external work to extract heat q_C (q_{in}) from the outside air (cold region) and deliver heat q_H (q_{out}) to the inside air (hot region). The most common heat pump is a phase-change heat pump. During the cycle of such a heat pump, the compressor pumps a gas through the condenser where it gets cooled down and finally condenses into liquid phase after releasing heat of condensation q_{out}. Then the liquid flows through an expansion valve where its pressure and temperature both drop considerably. Further it flows through the evaporator where it warms up and evaporates to gaseous phase again by extracting heat of vaporization q_{in} from the surroundings.

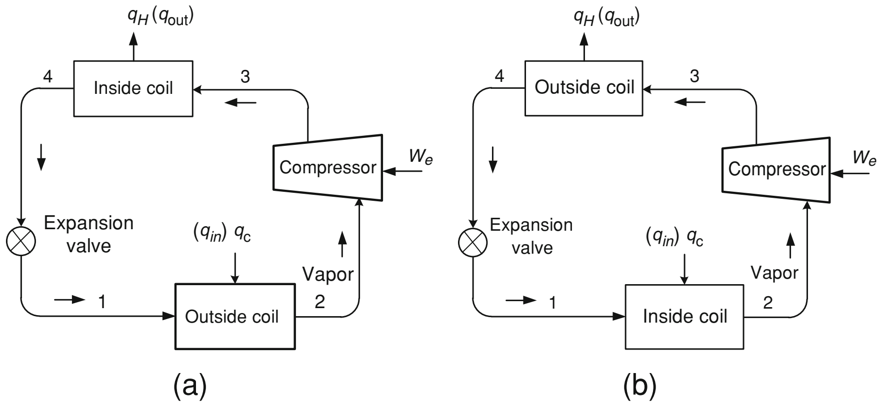


Fig. 7.15 A heat pump can be used: **a** to heat a house in the winter, and **b** to cool in the summer. A reversing valve reverses the direction of the fluid-flow so the inside coil in the summer is used as outside coil in the winter

In heat pumps, the work energy W_{in} provided mainly in the form of electricity is converted into heat, and the sum of this energy and the heat energy that is moved from the cold reservoir (q_C) is equal to the total heat energy added to the hot reservoir q_H

$$\dot{q}_H = \dot{q}_C + \dot{W}_{\text{in}} \tag{7.69}$$

Thermal efficiency of heat pumps are measured by the coefficient of performance COP_{HP} defined by

$$\text{COP}_{\text{HP}} = \frac{\dot{q}_H}{\dot{W}_{\text{in}}} \tag{7.70}$$

The amount of heat they move can be greater than the input work. Therefore, heat pumps can be a more efficient way of heating than simply converting the input work into heat, as in an electric heater or furnace. The limiting value of the Carnot efficiency for the heat pump is

$$\text{COP}_{\text{HP}} \leq \frac{T_H}{T_H - T_C} \quad \text{(heating)} \tag{7.71}$$

Equation 7.71 shows that the COP decreases with increasing temperature difference between the hot and cold regions.

Figure 7.15 compares the flow directions of the working fluid when the heat pump is used for heating and for cooling. A reversing valve reverses the direction of the fluid-flow so the inside coil in the summer is used as the outside coil in the winter. This means that the working fluid is evaporated in the inside coil extracting heat from the warm air inside and is condensed outside discharging heat to warm outside surroundings in the summer [6]. Example 7.23 illustrates the analysis of a heat pump.

Example 7.23 Heat pump calculations
A heat pump provides 60 MJ/h to a house. If the compressor requires an electrical energy input of 5 kW, calculate the COP. If electricity costs $0.08 per kWh and the heat pump operates 100 h per month, how much money does the homeowner save by using the heat pump instead of an electrical resistance heater?

Solution:
The heat pump operates at steady-state.
COP for a heat pump with a heat supply of 60 MJ/h = 16.66 kW: and W_{HP} = 5 kW

$$\mathrm{COP}_{\mathrm{HP}} = \frac{q_{\mathrm{out}}}{W_{\mathrm{HP}}} = \mathbf{3.33}$$

An electrical resistance heater converts all of the electrical work supplied W_e into heat q_H. Therefore, in order to get 16.66 kW into your home, you must buy 16.66 kW of electrical power.
Cost of resistance heater:
Power = 16.66 kW

$$\begin{aligned}\mathrm{Cost}(\$/\mathrm{month}) &= \mathrm{Electricity}(\$0.08/\mathrm{kWh})\mathrm{Power}(16.66\,\mathrm{kW})\mathrm{Time}(100\,\mathrm{h/month})\\ &= 133.3\,\$/\mathrm{month}\end{aligned}$$

Cost of heat pump with a power of 5 kW:

$$\begin{aligned}\mathrm{Cost}(\$/\mathrm{month}) &= \mathrm{Electricity}(\$0.08/\mathrm{kWh})\mathrm{Power}(5.0\,\mathrm{kW})\mathrm{Time}(100\,\mathrm{h/month})\\ &= 40.0\,\$/\mathrm{month}\end{aligned}$$

Therefore, monthly saving is (133.3 − 40.0) $/month = **93.3 $/month**.
Electrical resistance heaters are not very popular, especially in cold climates. The thermal efficiency of a heat pump drops significantly as the outside temperature falls. When the outside temperature drops far enough that the $\mathrm{COP}_{\mathrm{HP}} \sim 1$, it becomes more practical to use the resistance heater.

7.15.2 Refrigerators

Figure 7.16 shows a typical refrigeration cycle. A refrigerator is a heat engine in which work is required to extract energy from the freezing compartment and discharge that heat to the room through a condenser at the back of the refrigerator. This leads to further cooling of the cold region. One of the common refrigerant is 1,1,1,2-tetrafluoroethane (CF_3CH_2F) known as R-134a. R-134a has a boiling point temperature of –26.2°C (–15.0 °F) and a latent heat of 216.8 kJ/kg at 1.013 bar. It is compatible with most existing refrigeration equipment. R-134a has no harmful influence on the ozone layer of the earth's atmosphere. It is noncorrosive and nonflammable. R-134a is used for medium-temperature applications, such as air

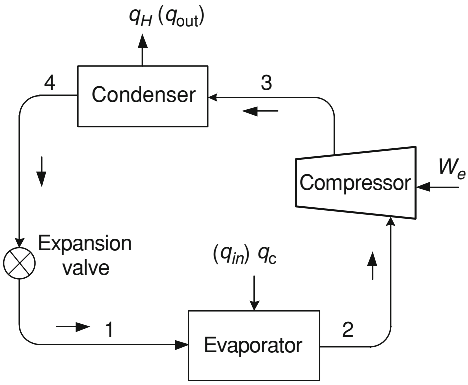


Fig. 7.16 A typical refrigeration cycle; the evaporator is the freezing compartment of the fridge, where the refrigerant absorb heat to be vaporized. The condenser is usually at the back of the fridge, where the heat of condensation is discharged to outside hot surroundings

conditioning and commercial refrigeration. Another refrigerant is pentafluoroethane (C_2HF_5) known as R-125, which is used in low- and medium-temperature applications. With a boiling point of −55.3°F at atmospheric pressure, R-125 is nontoxic, nonflammable, and noncorrosive.

Thermal efficiency of refrigerators and air conditioners is called the coefficient of performance COP_R:

$$COP_R = \frac{\dot{q}_C}{\dot{W}_{in}} \tag{7.72}$$

The limiting value of the Carnot efficiency for the refrigeration processes is

$$COP_R \leq \frac{T_C}{T_H - T_C} \quad \text{(cooling)} \tag{7.73}$$

Equation 7.73 shows that the COP_R decreases with increasing temperature difference between the hot and cold regions. Example 7.24 illustrates the analysis of a refrigerator. When the desired effect is cooling the heat resulting from the input work is just an unwanted byproduct. In everyday usage the efficiency of air conditioners is often rated by the *Seasonal Energy Efficiency Ratio* (SEER), which is discussed in detail in Sect. 9.4. Example 7.25 illustrates the estimation of the heat rejected in refrigeration cycle, while Example 7.26 illustrates the coefficient of performance estimation.

Example 7.24 Analysis of a refrigeration cycle

In a refrigeration cycle, the superheated R-134a (state 2) enters a compressor at 263.15 K and 0.18 MPa. The R-134a (state 3) leaves the compressor at 313.15 K and 0.6 MPa, and enters a condenser, where it is cooled by cooling water. The R-134a (state 4) leaves the condenser at 293.15 K and 0.57 MPa as saturated liquid, and enters a throttling valve. The partially vaporized R-134a (state 1) leaves the valve at 0.293 MPa. The cycle of R-134a is completed when it passes through an evaporator to absorb heat from the matter to be refrigerated. The flow rate of R-134a is 0.2 kg/s. The total power input is 60 kW. The surroundings are at 290 K. Determine the coefficient of performance and the exergy loss of the cycle.

Solution:
Assume that kinetic and potential energy changes are negligible, and the system is at steady-state.

Consider Fig. 7.16.
From Tables E1 to E2 the data for R-134a:
H_2 = 242.06 kJ/kg; at T_2 = 263.15 K; P_2 = 0.18 MPa (superheated vapor)

H_3 = 278.09 kJ/kg, at T_3 = 313.15 K, P_3 = 0.6 MPa (superheated vapor)

H_4 = 77.26 kJ/kg, at T_4 = 293.15 K, P_4 = 0.571 MPa (saturated liquid)
P_1 = 0.293 MPa, T_1 = 273.15 K

$W_{\text{in}} = 60\,\text{kW},\ \dot{m}_{\text{r}} = 0.2\,\text{kg/s}$

T_{o} = 290 K, $T_{\text{evaporator}}$ = 273 K, $T_{\text{condenser}}$ = 290 K

The throttling process where H_4 = H_1 = 77.26 kJ/kg (Stage 1) causes partial vaporization of the saturated liquid coming from the condenser.

The vapor part of the mixture, known as 'quality,' can be obtained using the enthalpy values at 0.293 MPa

$H_{1,\text{sat liq}}$ = 50.02 kJ/kg, $H_{1,\text{sat vap}}$ = 247.23 kJ/kg

$$x_1 = \frac{77.26 - 50.02}{247.23 - 50.02} = 0.138$$

For the cycle, the total enthalpy change is zero.

At the compressor, outside energy W_{in} is needed
At the evaporator, heat transfer, q_{in}, from the matter to be cooled is used to evaporate the refrigerant R-134a.

The heat absorbed within the evaporator from the contents of the refrigerator is

$$\dot{q}_{\text{in}} = \dot{m}_r(H_2 - H_1) = 32.96\,\text{kW}$$

The energy balance indicates the total energy (W_{in} + q_{in}) removed:

$$\dot{W}_{\text{in}} + \dot{q}_{\text{in}} = \dot{q}_{\text{out}} = 92.96\,\text{kW}$$

$$\dot{W}_{\text{ideal,in}} = \dot{m}_{\text{r}}(H_3 - H_2) = 7.20\,\text{kW}$$

Coefficient of performance (COP) of the refrigerator:
$$\text{COP} = \frac{\dot{q}_{\text{in}}}{\dot{W}_{\text{ideal,in}}} = \frac{H_2 - H_1}{H_3 - H_2} = \mathbf{4.57}$$
The total work (exergy) loss:
$$\dot{E}x_{\text{total}} = \dot{W}_{\text{in}} + \left(1 - \frac{T_0}{T_{\text{evaporater}}}\right)\dot{q}_{\text{in}} - \left(1 - \frac{T_0}{T_{\text{condenser}}}\right)(-\dot{q}_{\text{out}}) = \mathbf{58.94\,kW}$$

Exergy analysis identifies the performance of individual processes. Finding ways to improve the thermodynamic performance of individual steps is equally important.

Example 7.25 Heat rejection by a refrigerator
Food compartment of a refrigerator is maintained at 4°C by removing heat from it at a rate of 350 kJ/min. If the required power input of the refrigerator is 1.8 kW, determine (a) the coefficient of performance (COP) of the refrigerator, (b) the rate of heat discharged to the surroundings.

Solution:
Assume: steady-state operation.

$$\dot{q}_{\mathrm{c}} = 350\,\mathrm{kJ/min}, \quad \dot{W}_{\mathrm{net,in}} = 1.8\,\mathrm{kW},$$

$$\mathrm{COP_R} = \frac{\dot{q}_C}{\dot{W}_{\mathrm{net,in}}} = \frac{350\,\mathrm{kJ/min}}{1.8\,\mathrm{kW}}\left(\frac{\mathrm{kW}}{60\,\mathrm{kJ/min}}\right) = \mathbf{3.2}$$

This means that 3.2 kJ of heat is removed from the refrigerator for each kJ of energy supplied.

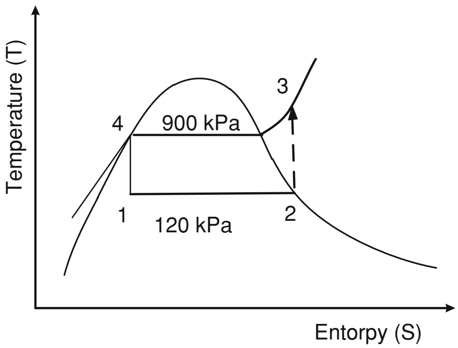
(b) Energy balance

$$\dot{q}_H = \dot{q}_C + \dot{W}_{\mathrm{net,in}} = 350\,\mathrm{kJ/min} + 1.8\,\mathrm{kW}\left(\frac{60\,\mathrm{kJ/min}}{kW}\right) = \mathbf{458\,kJ/min}$$

Both the heat removed from the refrigerator space and the energy supplied to the refrigerator as electrical work are discharged to surrounding air.

Example 7.26 Coefficient of performance of a vapor-compression refrigeration cycle
An ideal vapor-compression refrigeration cycle uses R-134a. The compressor inlet and outlet pressures are 120 and 900 kPa. The mass flow rate of refrigerant is 0.04 kg/s. Determine the coefficient of performance.

Solution;
Assume: steady-state operation. The changes in kinetic and potential energies are negligible.



Solution:
Assume: steady-state adiabatic operation. The changes in kinetic and potential energies are negligible.

The refrigerant mass flow rate = 0.04 kg/s
P_2 = 120 kPa, H_2 = 233.86 kJ/kg, S_2 = 0.9354 kJ/kg K
P_3 = 900 kPa, For the isentropic compression:
$S_3 = S_2$ = 0.9354 kJ/kg K → H_3 = 276.7 kJ/kg,
P_4 = 900 kPa, $H_{4\mathrm{sat\ liq}}$ = 99.56 kJ/kg; $H_{1\mathrm{sat\ liq}} = H_{4\mathrm{sat\ liq}}$ (throttling)

Power input to the compressor:
$\dot{W}_{\mathrm{in}} = \dot{m}(H_3 - H_2) = (0.04\,\mathrm{kg/s})(276.7 - 233.86)\,\mathrm{kJ/kg} = 1.71\,\mathrm{kW}$
Heat removed from the refrigerator space:

$$\dot{q}_C = \dot{m}(H_2 - H_1) = (0.04\,\mathrm{kg/s})(233.86 - 99.56)\,\mathrm{kJ/kg} = 5.37\,\mathrm{kW}$$

The heat discharged from the refrigerator to the surrounding:

$$\dot{q}_H = \dot{m}(H_3 - H_4) = (0.04\,\mathrm{kg/s})(276.7 - 99.56)\,\mathrm{kJ/kg} = 7.08\,\mathrm{kW}$$

$$\mathrm{COP}_R = \frac{\dot{q}_C}{\dot{W}_{\mathrm{in}}} = \frac{5.37\,\mathrm{kW}}{1.71\,\mathrm{kW}} = \mathbf{3.14}$$

This refrigerator is capable of removing 3.14 units of energy from the refrigerated space for each unit of electric energy it consumes.

7.16 Efficiency of Fuel Cells

A *fuel cell* is an electrochemical cell that converts chemical energy of a fuel into electric energy. Electricity is generated from the reaction between a fuel supply and an oxidizing agent. A hydrogen fuel cell uses hydrogen as fuel and oxygen (usually from air) as oxidant. Other fuels include hydrocarbons and alcohols (see Sect. 6.15). A typical fuel cell produces a voltage from 0.6 to 0.7 V at full rated load. To deliver the desired amount of energy, the fuel cells can be combined in series and parallel circuits, where series circuits yield higher voltage, and parallel circuits allow a higher current to be supplied. Such a design is called a *fuel cell stack*. The cell surface area can be increased to allow stronger current from each cell [22].

The efficiency of a fuel cell depends on the amount of power drawn from it. Drawing more power means drawing more current and hence increasing the losses in the fuel cell. Most losses appear as a voltage drop in the cell, so the efficiency of a cell is almost proportional to its voltage. For this reason, it is common to show graphs of voltage versus current (polarization curves) for fuel cells. A typical cell running at 0.7 V has an efficiency of about 50%, meaning that 50% of the energy content of the hydrogen is converted into electrical

Table 7.5 Types of fuel cell using polymer membrane as electrolyte

Fuel cell	Power output	T (°C)	Cell efficiency (%)
Proton exchange membrane fuel cell	100 W–500 kW	150–120 (Nafion)	50–70
Direct methanol fuel cell	100 mW–1 kW	90–120	20–30
Microbial fuel cell	Low	< 40	Low

Mench [26]; Vielstich et al. [39]

energy and the remaining 50% will be converted into heat. Fuel cells are not heat engines and so their efficiency is not limited by the Carnot cycle efficiency. Consequently, they can have very high efficiencies in converting chemical energy to electrical energy, especially when they are operated at low power density, and using pure hydrogen and oxygen as reactants. Fuel cell vehicles running on compressed hydrogen may have a power-plant-to-wheel efficiency of 22% if the hydrogen is stored as high-pressure gas [19, 39]. Table 7.5 compares the efficiency of several fuel cells.

The overall efficiency (electricity to hydrogen and back to electricity) of such plants (known as *round-trip efficiency*) is between 30 and 50%, depending on conditions. While a much cheaper lead-acid battery might return about 90%, the electrolyzer/fuel cell system can store indefinite quantities of hydrogen, and is therefore better suited for long-term storage.

7.17 Energy Conversions in Biological Systems

All living systems have to convert energy to the chemical energy in the form of energy rich chemical compounds. The two biochemical cyclic processes for such conversions are the oxidative phosphorylation in animals and the photosynthesis in plants. These cycles are discussed briefly in the next sections.

7.17.1 Energy Conversion by Oxidative Phosphorylation

In oxidative phosphorylation, the electrons are removed from food molecules in electron transport chain. A series of proteins in the membranes of mitochondria use the energy released from passing electrons from reduced molecules like NADH onto oxygen to pump protons across the membrane in mitochondria. The flow of protons causes the rotation of stalk subunit of a large protein called the ATPase. The rotation of ATPase changes the shape of the active site and synthesize adenosine triphosphate (ATP) from adenosine diphosphate (ADP) and inorganic phosphorus (Pi)

$$\mathrm{ADP} + \mathrm{Pi} = \mathrm{ATP} \tag{7.74}$$

ATP is an energy-rich compound having three phosphate group attached to a nucleoside of adenine called adenosine. Of the three phosphate groups, the terminal one has a weak linkage. This phosphate group can break spontaneously whenever ATP forms a complex with an enzyme. The breaking up of this bond releases chemical energy causing an immediate shift in the bond energy giving rise to ADP. The energy of ATP is used for all the activity of living cells, such as transport of ions and molecules, synthesis of new proteins and other substances, and the growth and development. ATP therefore acts as 'energy currency of the cell' and is used to transfer chemical energy between different biochemical reaction cycles [7, 25].

7.17.2 Energy from Photosynthesis

Photosynthesis is the synthesis of carbohydrates from sunlight and carbon dioxide (CO_2). The capture of solar energy is similar in principle to oxidative phosphorylation, as the proton motive force then drives ATP synthesis. The electrons needed to drive this electron transport chain come from light-gathering proteins called photosynthetic reaction centers. In plants, cyanobacteria and algae, oxygenic photosynthesis splits water, with oxygen produced as a waste product. This process uses the ATP and NADPH produced by the photosynthetic reaction centers. This carbon-fixation reaction is carried out by the enzyme RUBisCO.

7.17.3 Metabolism

Metabolism is the set of biochemical reactions that occur in living organisms to maintain life. Metabolism is usually divided into two categories catabolism and anabolism. Catabolism breaks down organic matter, for example to harvest energy in cellular respiration. Anabolism uses energy to construct components of cells such as proteins and nucleic acids. Adenosine triphosphate (ATP) is used to transfer chemical energy between different biochemical reaction cycles. ATP in cells is continuously regenerated and acts as a bridge between catabolism and anabolism, with catabolic reactions generating ATP and anabolic reactions consuming it.

7.17.4 Biological Fuels

Biological fuels can be categorized in three groups: carbohydrates (CH), representing a mixture of mono-, di-, and poly-saccarides, fats (F), and proteins (Pr) [25]. Carbohydrates are straight-chain aldehydes or ketones with many hydroxyl

groups that can exist as straight chains or rings. Carbohydrates are the most abundant biological molecules, and play numerous roles, such as the storage and transport of energy (starch, glycogen) and structural components such as cellulose in plants.

The fuel value is equal to the heat of reaction of combustion (oxidation). Carbohydrates and fats can be completely oxidized while proteins can only be partially oxidized and hence a lower fuel value. The energy expenditure may be calculated from the energy balance. Assume that (i) carbohydrate (CH), fat (F), and protein (Pr) are the only compounds involved in the oxidation process, (ii) the other compounds are stationary, and (iii) the uptake and elimination of oxygen, carbon dioxide, and nitrogen is instantaneous. Energy balance is

$$\dot{E} = \sum_i (\dot{n}\Delta H_r)_i = (\dot{n}\Delta H_r)_{\mathrm{CH}} + (\dot{n}\Delta H_r)_{\mathrm{F}} + (\dot{n}\Delta H_r)_{\mathrm{Pr}} = \dot{q} + \dot{W} \tag{7.75}$$

7.17.5 Converting Biomass to Biofuels

Technologies for converting biomass to biofuels are often classified in two main categories: *thermochemical conversion* and *biological conversion*.

- *Thermochemical conversion* involves applying heat to break down biomass into chemical intermediates that can be used to make fuel substitutes. Many of these thermal technologies are known for well over a century and are used primarily in transforming coal into fuels. Gasification combined with catalytic conversion of syngas (carbon monoxide and hydrogen) to fuels, for example, hydrogen, liquid fuels, mixed alcohols, or dimethyl ether. Pyrolysis of biomass produces bio-oils that could serve as intermediates in a petroleum refinery.
- *Biological conversion* focuses on fermentation of carbohydrates in biomass to ethanol and other chemicals. Fermentation technology, of course, is among the earliest conversion processes [38]. Optimal combinations of both biological and thermochemical fuel production may lead to greater energy efficiency in the transformation of the energy of biomass into useable fuels.

In 2005, 3.9 billion gallons of ethanol fuel primarily from the fermentation of starch in corn grain were produced and sold in the United States. Estimations show that an upper limit on ethanol production from corn at around 10 billion gallons per year, which represents a very small fraction of U.S. demand for gasoline [23, 34]. Because a gallon of ethanol contains only 67% as much energy as a gallon of gasoline, 10 billion gallons of ethanol would represent only 7 billion gallons per year corresponding to 5% of gasoline demand in U.S.

Accessing energy in lignocellulosic biomass greatly increases the potential supply of biofuels. Figure 7.17 provides a schematic representation of the conversion process of cellulosic feedstock to biofuels. Cellulosic biomass needs

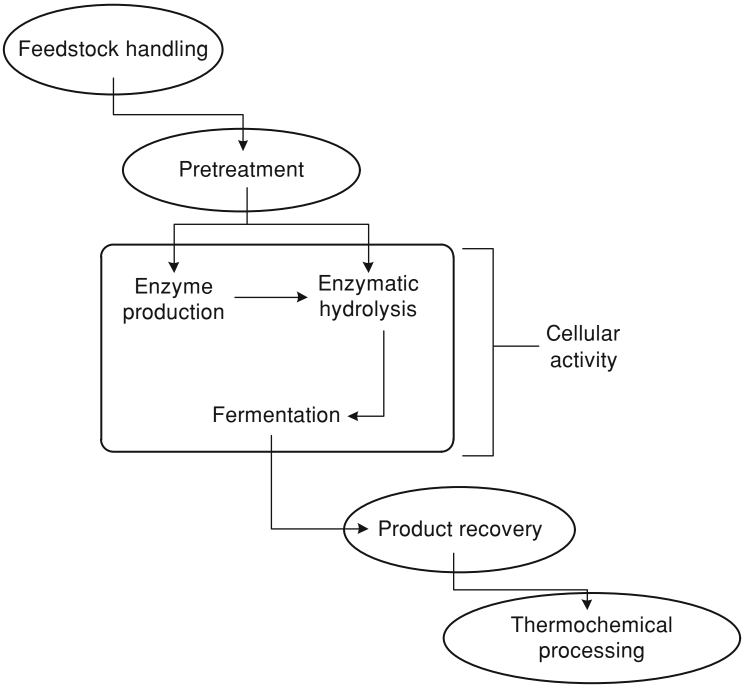


Fig. 7.17 Process schematic of biological conversion of lignocellulosic biomass to ethanol

pretreatment and enzymatic processes before it can be converted to biofuel by fermentation. The critical technology advancements need of biological catalysts that can cost-effectively break down the carbohydrate polymers in biomass to sugars and microbes that can ferment all of the sugars in biomass to ethanol [9, 11, 30].

Government incentives for biodiesel have spurred significant growth in the vegetable oil-based biodiesel fuel substitute. Converting vegetable oil to biodiesel is a relatively simple thermochemical process in which the natural oil is synthesized with methanol to form fatty acid methyl ester [23].

Problems

7.1 A pump delivers water from the bottom of a storage tank open to the atmosphere containing water at 75°F. The water in the storage tank is 10 ft deep and bottom of the tank is 50 ft above the ground. The pump delivers the water at 80°F and 100 psia to another tank, which is 10 ft above the ground. If the water flow rate is 5,000 lb/h and pump efficiency is 80%, estimate the power in hp required.

7.2 The pump of a water storage tank is powered with a 18-kW electric motor operating with an efficiency of 90%. The water flow rate is 40 l/s. The diameters of the inlet and exit pipes are the same, and the elevation difference between the inlet and outlet is negligible. The absolute pressures at the inlet and outlet are 100 and 400 kPa, respectively. Determine the mechanical efficiency of the pump.

7.3 The pump of a water storage tank is powered with a 18-kW electric motor operating with an efficiency of 85%. The water flow rate is 30 l/s. The diameters of the inlet and exit pipes are the same, and the elevation difference between the inlet and outlet is negligible. The absolute pressures at the inlet and outlet are 100 and 500 kPa, respectively. Determine the mechanical efficiency of the pump.

7.4 A wind turbine-generator with a 24-foot-diameter blade produces 0.35 kW of electric power. In the location of the wind turbine, the wind speed is 11 mile per hour. Determine the efficiency of the wind turbine-generator.

7.5 A wind turbine-generator with a 27-foot-diameter blade produces 0.4 kW of electric power. In the location of the wind turbine, the wind speed is 10 mile per hour. Determine the efficiency of the wind turbine-generator.

7.6 A hydroelectric plant operates by water falling from a 25 m height. The turbine in the plant converts potential energy into electrical energy with an assumed efficiency of 85%. The electricity is lost by about 8% through the power transmission. Estimate the mass flow rate of the water necessary to power a 500 W light bulb.

7.7 A hydroelectric plant is generating 125 kW of power. If their turbines only convert the potential energy form the water with a 75% efficiency, find the height change for the water flow rate of 480 kg/s must fall in order to continue to make 125 kW of power.

7.8 A hydroelectric plant operates by water falling from a 40 m height. The turbine in the plant converts potential energy into electrical energy with an assumed efficiency of 80%. The electricity is lost by about 8% through the power transmission. Estimate the mass flow rate of the water necessary to power a 1,500 W light bulb.

7.9 A hydroelectric plant operates by water falling from a 150 ft height. The turbine in the plant converts potential energy into electrical energy with an assumed efficiency of 82%. The electricity is lost by about 6% through the power transmission. Estimate the mass flow rate of the water necessary to power a 3,500 W light bulb.

7.10 A hydroelectric plant operates by water falling from a 200 ft height. The turbine in the plant converts potential energy into electrical energy with an assumed efficiency of 85%. The electricity is lost by about 5% through the power transmission so the available power is 95%. Estimate the mass flow rate of the water necessary to power a 3,500 W light bulb.

7.11 A hydroelectric plant operates by water falling from a 200 ft height. The turbine in the plant converts potential energy into electrical energy with an assumed efficiency of 85%. The power is lost by about 5% through the

power transmission so the available power is 95%. If the mass flow rate of the water 396 lb/s, estimate the power output of the hydro plant.

7.12 An electric motor attached to a pump draws 10.2 A at 110 V. At steady load the motor delivers 1.32 hp of mechanical energy. Estimate the rate of heat transfer from the motor.

7.13 An electric motor attached to a pump draws 12 A at 110 V. At steady load the motor delivers 1.5 hp of mechanical energy. Estimate the rate of heat transfer from the motor.

7.14 Air enters an insulated compressor operating at steady-state at 1.0 bar, 300 K with a mass flow rate of 3.6 kg/s and exits at 2.76 bar. Kinetic and potential energy effects are negligible. (a) Determine the minimum theoretical power input required, in kW, and the corresponding exit temperature, in K. (b) If the actual exit temperature is 420 K, determine the power input, in kW, and the isentropic compressor efficiency.

7.15 Air enters an insulated compressor operating at steady-state at 1.0 bar, 300 K with a mass flow rate of 2.5 kg/s and exits at 2.6 bar. Kinetic and potential energy effects are negligible. (a) Determine the minimum theoretical power input required, in kW, and the corresponding exit temperature, in K. (b) If the actual exit temperature is 420 K, determine the power input, in kW, and the isentropic compressor efficiency.

7.16 A compressor increases the pressure of carbon dioxide from 100 to 600 kPa. The inlet temperature is 300 K and the outlet temperature is 400 K. The mass flow rate of carbon dioxide is 0.01 kmol/s. The power required by the compressor is 55 kW. The temperature of the surroundings is 290 K. Determine the minimum amount of work required and the coefficient of performance.

7.17 A compressor increases the pressure of carbon dioxide from 100 to 500 kPa. The inlet temperature is 300 K and the outlet temperature is 400 K. The mass flow rate of carbon dioxide is 0.015 kmol/s. The power required by the compressor is 60 kW. The temperature of the surroundings is 290 K. Determine the minimum amount of work required and the coefficient of performance.

7.18 A compressor receives air at 15 psia and 80°F with a flow rate of 1.2 lb/s. The air exits at 40 psia and 300°F. At the inlet the air velocity is low, but increases to 250 ft/s at the outlet of the compressor. Estimate the power input to the compressor if it is cooled at a rate of 200 Btu/s.

7.19 A compressor receives air at 15 psia and 80°F with a flow rate of 1.5 lb/s. The air exits at 50 psia and 300°F. At the inlet the air velocity is low, but increases to 250 ft/s at the outlet of the compressor. Estimate the power input to the compressor if it is cooled at a rate of 150 Btu/s.

7.20 In an adiabatic compression operation, air is compressed from 20°C and 101.32 to 520 kPa with an efficiency of 0.7. The air flow rate is 22 mol/s. Assume that the air remains ideal-gas during the compression. The surroundings are at 298.15 K. Determine the thermodynamic efficiency η_{th} and

the rate of energy dissipated $\dot{E}_{loss}$.

7.21 In an adiabatic compression operation, air is compressed from 25°C and 101.32 to 560 kPa with an efficiency of 0.7. The air flow rate is 20 mol/s. Assume that the air remains ideal-gas during the compression. The surroundings are at 298.15 K. Determine the thermodynamic efficiency η_{th} and the rate of energy dissipated $\dot{E}_{loss}$.

7.22 The power required to compress 0.05 kg/s of steam from a saturated vapor state at 50°C to a pressure of 800 kPa at 200°C is 15 kW. Find the conversion rate of power input to heat loss from the compressor.

7.23 The power required to compress 0.1 kg/s of steam from a saturated vapor state at 80°C to a pressure of 1,000 kPa at 200°C is 28 kW. Find the conversion rate of power input to heat loss from the compressor.

7.24 A steam turbine consumes 4,000 lb/h steam at 540 psia and 800°F. The exhausted steam is at 165 psia. The turbine operation is adiabatic.
(a) Determine the exit temperature of the steam and the work produced by the turbine.
(b) Determine the thermal efficiency.

7.25 A steam turbine consumes 3,800 lb/h steam at 540 psia and 800°F. The exhausted steam is at 180 psia. The turbine operation is adiabatic.
(a) Determine the exit temperature of the steam and the work produced by the turbine.
(b) Determine the thermal efficiency.

7.26 A superheated steam (stream 1) expands in a turbine from 5,000 kPa and 325°C to 150 kPa and 200°C. The steam flow rate is 15.5 kg/s. If the turbine generates 1.1 MW of power, determine the heat loss to the surroundings and thermal efficiency.

7.27 A superheated steam (stream 1) expands in a turbine from 6,000 kPa and 325°C to 150 kPa and 200°C. The steam flow rate is 16.2 kg/s. If the turbine generates 1.2 MW of power, determine the heat loss to the surroundings and thermal efficiency.

7.28 Steam expands in a turbine from 6,600 kPa and 300°C to a saturated vapor at 1 atm. The steam flow rate is 9.55 kg/s. If the turbine generates a power of 1 MW, determine the thermal efficiency.

7.29 Steam expands in a turbine from 7,000 kPa and 300°C to a saturated vapor at 1 atm. The steam flow rate is 14.55 kg/s. If the turbine generates a power of 1.5 MW, determine the thermal efficiency.

7.30 Steam expands adiabatically in a turbine from 850 psia and 600°F to a wet vapor at 12 psia with a quality of 0.9. The turbine produces a power output of 1,500 Btu/s. Estimate the thermal efficiency for a steam flow rate of 12.8 lb/s.

7.31 Steam expands adiabatically in a turbine from 900 psia and 600°F to a wet vapor at 10 psia with a quality of 0.9. The turbine produces a power output of 1,550 Btu/s. Estimate the thermal efficiency for a steam flow rate of 14.2 lb/s.

7.32 A turbine produces 65,000 kW electricity with an efficiency of 70%. It uses a superheated steam at 8,200 kPa and 550°C. The discharged stream is a

saturated mixture at 75 kPa. If the expansion in the turbine is adiabatic, and the surroundings are at 298.15 K, determine the thermodynamic efficiency and the work loss.

7.33 A turbine produces 70 MW electricity with an efficiency of 70%. It uses a superheated steam at 8,800 kPa and 550°C. The discharged stream is a saturated mixture at 15 kPa. If the expansion in the turbine is adiabatic, and the surroundings are at 298.15 K, determine the thermodynamic efficiency and the work loss.

7.34 A Carnot cycle use water as the working fluid at a steady-flow process. Heat is transferred from a source at 250°C and water changes from saturated liquid to saturated vapor. The saturated steam expands in a turbine at 10 kPa, and heat is transferred in a condenser at 10 kPa. Estimate the thermal efficiency and net power output of the cycle.

7.35 A Carnot cycle use water as the working fluid at a steady-flow process. Heat is transferred from a source at 200°C and water changes from saturated liquid to saturated vapor. The saturated steam expands in a turbine at 20 kPa, and heat is transferred in a condenser at 20 kPa. Estimate the thermal efficiency of the cycle and the amount of heat transferred in the condenser for a flow rate of 5.5 kg/s of the working fluid.

7.36 A Carnot cycle use water as the working fluid at a steady-flow process. Heat is transferred from a source at 200°C and water changes from saturated liquid to saturated vapor. The saturated steam expands in a turbine at 10 kPa, and heat is transferred in a condenser at 10 kPa. Estimate the thermal efficiency of the cycle and the amount of heat transferred in the condenser for a flow rate of 7.5 kg/s of the working fluid.

7.37 A Carnot cycle use water as the working fluid at a steady-flow process. Heat is transferred from a source at 400°F and water changes from saturated liquid to saturated vapor. The saturated steam expands in a turbine at 5 psia, and heat is transferred in a condenser at 5 psia. Estimate the thermal efficiency of the cycle and the amount of heat transferred in the condenser for a flow rate of 10 lb/s of the working fluid.

7.38 Consider a simple ideal Rankine cycle. If the turbine inlet temperature and the condenser pressure are kept the same, discuss the effects of increasing the boiler pressure on:

- Turbine power output
- Heat Supplied
- Thermal efficiency
- Heat rejected

7.39 Consider a simple ideal Rankine cycle. If the boiler and the condenser pressures are kept the same, discuss the effects of increasing the temperature of the superheated steam on:

- Turbine power output
- Heat Supplied

- Thermal efficiency
- Heat rejected

7.40 A steam power production plant burns fuel at 1,273.15 K (T_H), and cooling water is available at 290 K (T_C). The steam produced by the boiler is at 8,200 kPa and 823.15 K. The condenser produces a saturated liquid at 30 kPa. The turbine and pump operate reversibly and adiabatically. Determine the thermal efficiency of the cycle for the steam flow rate of 1 k/s.

7.41 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 3,000 kPa and 350°C. The condenser operates at 30 kPa. The mass flow rate of steam is 22 kg/s. Estimate the thermal efficiency of the cycle and the net power output.

7.42 A steam power plant operates on a simple ideal Rankine cycle shown below. The turbine receives steam at 698.15 K and 4,100 kPa, while the discharged steam is at 40 kPa. The mass flow rate of steam is 3.0 kg/s. In the boiler, heat is transferred into the steam from a source at 1,500 K (T_H). In the condenser, heat is discharged to the surroundings at 298 K (T_C) The condenser operates at 298 K. Determine the thermal efficiency of the cycle.

7.43 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 5,000 kPa and 300°C. The condenser operates at 20 kPa. The mass flow rate of steam is 25 kg/s. Estimate the thermal efficiency of the cycle and the net power output.

7.44 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 10,000 kPa and 400°C. The condenser operates at 30 kPa. The power output of the cycle is 140 MW. Estimate the thermal efficiency of the cycle and the mass flow rate of the steam.

7.45 A steam power plant is operating on the simple ideal Rankine cycle. The steam mass flow rate is 20 kg/s. The steam enters the turbine at 3,500 kPa and 400°C. Discharge pressure of the steam from the turbine is 15 kPa. Determine the thermal efficiency of the cycle.

7.46 A simple ideal Rankine cycle is used in a steam power plant. Steam enters the turbine at 6,600 kPa and 798.15 K. The net power output of the turbine is 35 kW. The discharged steam is at 10 kPa. Determine the thermal efficiency.

7.47 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 10,000 kPa and 500°C. The condenser operates at 10 kPa. The power output of the cycle is 175 MW. Turbine operates with an isentropic efficiency of 0.80, while the pump operates with an isentropic efficiency of 0.90. Estimate the thermal efficiency of the cycle and the heat transferred in the condenser.

7.48 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 10,000 kPa and 400°C. The condenser operates at 10 kPa. The power output of the cycle is 145 MW. Turbine operates with an isentropic efficiency of 0.85, while the pump operates with an isentropic efficiency of

0.95. Estimate the thermal efficiency of the cycle and the heat transferred in the condenser.

7.49 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 10,000 kPa and 400°C. The condenser operates at 10 kPa. The mass flow rate of steam is 110 kg/s. Turbine operates with an isentropic efficiency of 0.85, while the pump operates with an isentropic efficiency of 0.95. Estimate the thermal efficiency and the power output of the cycle.

7.50 A steam power plant operates on a simple ideal Rankine cycle. The boiler operates at 8,000 kPa and 400°C. The condenser operates at 20 kPa. The mass flow rate of steam is 80 kg/s. Turbine operates with an isentropic efficiency of 0.85, while the pump operates with an isentropic efficiency of 0.90. Estimate the thermal efficiency and the power output of the cycle.

7.51 A steam power plant shown below uses natural gas to produce 0.1 MW power. The combustion heat supplied to a boiler produces steam at 10,000 kPa and 798.15 K. The turbine efficiency is 0.7. The discharged steam from the turbine is at 30 kPa, and is sent to a condenser. The condensed water is pumped to the boiler. The pump efficiency is 0.90. Determine the thermal efficiency of the cycle.

7.52 A steam power plant shown below uses natural gas to produce 0.12 MW power. The combustion heat supplied to a boiler produces steam at 10,000 kPa and 798.15 K. The turbine efficiency is 0.75. The discharged steam from the turbine is at 30 kPa, and is sent to a condenser. The condensed water is pumped to the boiler. The pump efficiency is 0.85. Determine:
(a) The thermal efficiency of an ideal Rankine cycle.
(b) The thermal efficiency of an actual Rankine cycle.

7.53 A steam power plant shown below uses natural gas to produce 0.12 MW power. The combustion heat supplied to a boiler produces steam at 9,000 kPa and 798.15 K. The turbine efficiency is 0.8. The discharged steam from the turbine is at 10 kPa, and is sent to a condenser. The condensed water is pumped to the boiler. The pump efficiency is 0.9. Determine:
(a) The thermal efficiency of an ideal Rankine cycle.
(b) The thermal efficiency of an actual Rankine cycle.

7.54 A simple ideal reheat Rankine cycle is used in a steam power plant shown below. Steam enters the turbine at 9,000 kPa and 823.15 K and leaves at 4,350 kPa and 698.15 K. The steam is reheated at constant pressure to 823.15 K. The discharged steam from the low-pressure turbine is at 10 kPa. The net power output of the turbine is 40 MW. In the boiler, heat is transferred into the steam from a source at 1,600 K (T_H). In the condenser, heat is discharged to the surroundings at 298 K (T_C). The condenser operates at 298 K. The turbine efficiency is 0.8 and the pump efficiency is 0.9. Determine the mass flow rate of steam and the thermal efficiency of the cycle.

7.55 A steam power plant is using an ideal regenerative Rankine cycle shown below. Steam enters the high-pressure turbine at 8,200 kPa and 773.15 K, and the condenser operates at 20 kPa. The steam is extracted from the

turbine at 350 kPa to heat the feed water in an open heater. The water is a saturated liquid after passing through the feed water heater. The work output of the turbine is 50 MW. In the boiler, heat is transferred into the steam from a source at 1,600 K (T_H). In the condenser, heat is discharged to the surroundings at 285 K (T_C). Determine the thermal efficiency of the cycle.

7.56 A steam power plant is using an ideal reheat regenerative Rankine cycle shown below. Steam enters the high-pressure turbine at 9,000 kPa and 773.15 K and leaves at 850 kPa. The condenser operates at 10 kPa. Part of the steam is extracted from the turbine at 850 kPa to heat the water in an open heater, where the steam and liquid water from the condenser mix and direct contact heat transfer takes place. The rest of the steam is reheated to 723.15 K, and expanded in the low-pressure turbine section to the condenser pressure. The water is a saturated liquid after passing through the water heater and is at the heater pressure. The work output of the turbine is 75 MW. In the boiler, heat is transferred into the steam from a source at 1,600 K (T_H). In the condenser, heat is discharged to the surroundings at 285 K (T_C). Determine the thermal efficiency of the cycle.

7.57 A steam power plant is using a geothermal energy source. The geothermal source is available at 220°C and 2,320 kPa with a flow rate of 200 kg/s. The hot water goes through a valve and a flash drum. Steam from the flash drum enters the turbine at 550 kPa and 428.62 K. The discharged steam from the turbine has a quality of $x_4 = 0.96$. The condenser operates at 10 kPa. The water is a saturated liquid after passing through the condenser. Determine the thermal efficiency of the cycle.

7.58 A geothermal power production plant produces 7 MW power. Inlet temperature of the hot geothermal liquid source is 150°C. The flow rate of the hot liquid water is 220 kg/s. The reference state is at 25°C. Estimate the second law efficiency of the plant.

7.59 A reheat Rankine cycle is used in a steam power plant. Steam enters the high-pressure turbine at 9,000 kPa and 823.15 K and leaves at 4,350 kPa. The steam is reheated at constant pressure to 823.15 K. The steam enters the low-pressure turbine at 4,350 kPa and 823.15 K. The discharged steam from the low-pressure turbine is at 10 kPa. The net power output of the turbine is 65 MW. The isentropic turbine efficiency is 80%. The pump efficiency is 95%. In the boiler, heat is transferred into the steam from a source at 1,600 K. In the condenser, heat is discharged to the surroundings at 298 K. The condenser operates at 298 K. Determine the thermal efficiency.

7.60 A reheat Rankine cycle is used in a steam power plant. Steam enters the high-pressure turbine at 10,000 kPa and 823.15 K and leaves at 4,350 kPa. The steam is reheated at constant pressure to 823.15 K. The steam enters the low-pressure turbine at 4,350 kPa and 823.15 K. The discharged steam from the low-pressure turbine is at 15 kPa. The net power output of the turbine is 65 MW. The isentropic turbine efficiency is 80%. The pump efficiency is

95%. In the boiler, heat is transferred into the steam from a source at 1,600 K. The condenser operates at 298 K. Determine the thermal efficiency.

7.61 A steam power plant is using an actual regenerative Rankine cycle. Steam enters the high-pressure turbine at 11,000 kPa and 773.15 K, and the condenser operates at 10 kPa. The steam is extracted from the turbine at 475 kPa to heat the water in an open heater. The water is a saturated liquid after passing through the water heater. The work output of the turbine is 90 MW. The pump efficiency is 95% and the turbine efficiency is 75%. In the boiler, heat is transferred into the steam from a source at 1,700 K. In the condenser, heat is discharged to the surroundings at 285 K. Determine the thermal efficiency.

7.62 A steam power plant is using an actual regenerative Rankine cycle. Steam enters the high-pressure turbine at 10,000 kPa and 773.15 K, and the condenser operates at 15 kPa. The steam is extracted from the turbine at 475 kPa to heat the water in an open heater. The water is a saturated liquid after passing through the water heater. The work output of the turbine is 90 MW. The pump efficiency is 90% and the turbine efficiency is 80%. In the boiler, heat is transferred into the steam from a source at 1,700 K. In the condenser, heat is discharged to the surroundings at 285 K. Determine the thermal efficiency.

7.63 A steam power plant is using an actual reheat regenerative Rankine cycle. Steam enters the high-pressure turbine at 11,000 kPa and 773.15 K, and the condenser operates at 10 kPa. The steam is extracted from the turbine at 2,000 kPa to heat the water in an open heater. The steam is extracted at 475 kPa for process heat. The water is a saturated liquid after passing through the water heater. The work output of the turbine is 90 MW. The turbine efficiency is 80%. The pumps operate isentropically. In the boiler, heat is transferred into the steam from a source at 1,700 K. In the condenser, heat is discharged to the surroundings at 290 K. Determine the thermal efficiency of the plant.

7.64 A steam power plant is using an actual reheat regenerative Rankine cycle. Steam enters the high-pressure turbine at 10,000 kPa and 773.15 K, and the condenser operates at 15 kPa. The steam is extracted from the turbine at 2,000 kPa to heat the water in an open heater. The steam is extracted at 475 kPa for process heat. The water is a saturated liquid after passing through the water heater. The work output of the turbine is 85 MW. The turbine efficiency is 80%. The pumps operate isentropically. In the boiler, heat is transferred into the steam from a source at 1,700 K. In the condenser, heat is discharged to the surroundings at 290 K. Determine the thermal efficiency of the plant.

7.65 A steam power plant operates on a regenerative cycle. Steam enters the turbine at 700 psia and 800°F and expands to 1 psia in the condenser. Part of the steam is extracted at 60 psia. The efficiencies of the turbine and pump are 0.80 and 0.95, respectively. If the mass flow rate of steam is 9.75 lb/s estimate the thermal efficiency of the turbine.

7.66 A steam power plant operates on a regenerative cycle. Steam enters the turbine at 750 psia and 800°F and expands to 5 psia in the condenser. Part of the steam is extracted at 60 psia. The efficiencies of the turbine and pump are 0.80 and 0.90, respectively. If the mass flow rate of steam is 10.5 lb/s estimate the thermal efficiency of the turbine.

7.67 A cogeneration plant shown below uses steam at 900 psia and 1,000°F to produce power and process heat. The steam flow rate from the boiler is 16 lb/s. The process requires steam at 70 psia at a rate of 3.2 lb/s supplied by the expanding steam in the turbine. The extracted steam is condensed and mixed with the water output of the condenser. The remaining steam expands from 70 psia to the condenser pressure of 3.2 psia. In the boiler, heat is transferred into the steam from a source at 3,000 R. In the condenser, heat is discharged to the surroundings at 540 R. If the turbine operates with an efficiency of 80% and the pumps with an efficiency of 85%, determine the thermal efficiency of the cycle.

7.68 A power plant is operating on an ideal Brayton cycle with a pressure ratio of $r_p = 9$. The fresh air temperature is 300 K at the compressor inlet and 1,200 K at the end of the compressor and at the inlet of the turbine. Using the standard-air assumptions, determine the thermal efficiency of the cycle.

7.69 A power plant is operating on an ideal Brayton cycle with a pressure ratio of $r_p = 9$. The fresh air temperature is 300 K at the compressor inlet and 1,200 K at the end of the compressor and at the inlet of the turbine. Assume the gas-turbine cycle operates with a compressor efficiency of 80% and a turbine efficiency of 80%. Determine the thermal efficiency of the cycle.

7.70 The net work of a power cycle is 8×10^6 Btu/s and the heat transfer to the cold reservoir, q_C, is 12×10^6 Btu/s. The hot source operates at 1,400 R and the cold source temperature is 560 R. What is the ratio of the achieved thermal efficiency to the maximum thermal efficiency of the cycle?

7.71 The net work of a power cycle is 25 MW and the heat transfer to the cold reservoir, q_C, is 67 MW. The hot source operates at 573 K and the cold source temperature is 285. What is the ratio of the achieved thermal efficiency to the maximum thermal efficiency of the cycle?

7.72 The net work of a power cycle is 52,000 Btu/s and the heat transfer to the cold reservoir, q_C, is 69,000 Btu/s. The hot source operates at 1,450 R and the cold source temperature is 550 R. What is the ratio of the achieved thermal efficiency to the maximum thermal efficiency of the cycle?

7.73 The net work of a power cycle is 75,000 Btu/s and the heat transfer to the cold reservoir, q_C, is 85,000 Btu/s. The hot source operates at 1,500 R and the cold source temperature is 530 R. What is the ratio of the achieved thermal efficiency to the maximum thermal efficiency of the cycle?

7.74 An ideal Otto cycle operates with a compression ratio (V_{max}/V_{min}) of 8.9. Air is at 101.3 kPa and 300 K at the start of compression (state 1). The maximum and minimum temperatures in the cycle are 1,360 and 300 K. Specific heats depend on the temperature. Determine the thermal efficiency

of the cycle and the thermal efficiency of a Carnot cycle working between the same temperature limits.

7.75 An ideal Otto cycle operates with a compression ratio (V_{max}/V_{min}) of 9.2. Air is at 101.3 kPa and 300 K at the start of compression (state 1). During the constant-volume heat-addition process, 730 kJ/kg of heat is transferred into the air from a source at 1,900 K. Heat is discharged to the surroundings at 280 K. Determine the thermal efficiency of energy conversion.

7.76 An ideal Otto cycle operates with a compression ratio (V_{max}/V_{min}) of 9. Air is at 101.3 kPa and 295 K at the start of compression (state 1). During the constant-volume heat-addition process, 900 kJ/kg of heat is transferred into the air from a source at 1,800 K. Heat is discharged to the surroundings at 295 K. Determine the thermal efficiency of the energy conversion.

7.77 An ideal Otto cycle operates with a compression ratio (= V_{max}/V_{min}) of 8.5. Air is at 101.3 kPa and 285 K at the start of compression (state 1). During the constant-volume heat-addition process, 1,000 kJ/kg of heat is transferred into the air from a source at 1,800 K. Heat is discharged to the surroundings at 280 K. Determine the thermal efficiency of energy conversion.

7.78 An ideal Otto cycle operates with a compression ratio (=V_{max}/V_{min}) of 10. Air is at 101.3 kPa and 295 K at the start of compression (state 1). During the constant-volume heat-addition process, 1,000 kJ/kg of heat is transferred into the air from a source at 1,800 K. Heat is discharged to the surroundings at 295 K. Determine the thermal efficiency of the energy conversion.

7.79 An ideal Otto cycle operates with a compression ratio (r = V_{max}/V_{min}) of 8. Air is at 101.3 kPa and 300 K at the start of compression (state 1). The maximum and minimum temperatures in the cycle are 1,300 and 300 K, respectively. Determine the thermal efficiency of energy conversion and the thermal efficiency of the Carnot engine. The average specific heats are $C_{p,av}$ = 1.00 kJ/kg K and $C_{v,av}$ = 0.717 kJ/kg K

7.80 An ideal Otto cycle operates with a compression ratio (r = V_{max}/V_{min}) of 8.8. Air is at 101.3 kPa and 300 K at the start of compression (state 1). The maximum and minimum temperatures in the cycle are 1,400 and 300 K, respectively.. Determine the thermal efficiency of energy conversion and the thermal efficiency of the Carnot engine. The average specific heats are $C_{p,av}$ = 1.00 kJ/kg K and $C_{v,av}$ = 0.717 kJ/kg K

7.81 One kmol of carbon dioxide is initially at 1 atm and −13°C performs a power cycle consisting of three internally reversible processes in series. Step 1–2: adiabatic compression to 5 atm. Step 2–3: isothermal expansion to 1 atm. Step 3–1: constant-pressure compression. Determine the net work, in Btu/lb_m and the thermal efficiency.

7.82 An ideal Diesel cycle has an air-compression ratio of 18 and operating with maximum temperature of 2,660 R. At the beginning of the compression, the fluid pressure and temperature are 14.7 psia, 540, respectively. The average specific heats of air at room temperature are $C_{p,av}$ = 0.24 Btu/lb R and, $C_{v,av}$

= 0.171 Btu/lb R. Utilizing the cold-air-standard assumptions, determine the thermal efficiency.

7.83 An ideal Diesel cycle has an air-compression ratio of 18 and operating with maximum temperature of 2,200 K. At the beginning of the compression, the fluid pressure and temperature are 100 kPa, and 290 K, respectively. The average specific heats of air at room temperature are $C_{p,av}$ = 1.0 kJ/kg K and, $C_{v,av}$ = 0. 718 kJ/kg K. Utilizing the cold-air-standard assumptions, determine the thermal efficiency of the cycle.

7.84 An ideal Diesel cycle has an air-compression ratio of 16 and a cutoff ratio of 2. At the beginning of the compression, the fluid pressure, temperature, and volume are 100 kPa, 300 K, respectively. The average specific heats of air at room temperature are $C_{p,av}$ = 1.005 kJ/kg K, $C_{v,av}$ = 0.7181 kJ/kg K. Utilizing the cold-air-standard assumptions, determine the thermal efficiency of the cycle.

7.85 An ideal Diesel cycle has an air-compression ratio of 16 and a cutoff ratio of 2. At the beginning of the compression, the fluid pressure, temperature, and volume are 100 kPa, 290 K, respectively. The average specific heats of air at room temperature are $C_{p,av}$ = 1.005 kJ/kg K, $C_{v,av}$ = 0.7181 kJ/kg K. Utilizing the cold-air-standard assumptions, determine the thermal efficiency of the cycle.

7.86 A refrigerator using tetrafluoroethane (R-134a) as refrigerant operates with a capacity of 10,000 Btu/h. The refrigerated space is at 15°F. The evaporator and condenser operate with a 10°F temperature difference in their heat transfer. Cooling water enters the condenser at 70°F. Therefore, the evaporator is at 5°F, and the condenser is at 80°F. Determine the ideal and actual power necessary if the compressor efficiency is 85%.

7.87 In a refrigeration cycle, the superheated R-134a (state 2) enters a compressor at 200°F and 90.0 psia. The R-134a (state 3) leaves the compressor at 360°F and 140 psia, and enters a condenser, where it is cooled by cooling water. The R-134a (state 4) leaves the condenser at 90.5°F and 120 psia as saturated liquid, and enters a throttling valve. The partially vaporized R-134a (state 1) leaves the valve at 100 psia. The cycle of R-134a is completed when it passes through an evaporator to absorb heat from the matter to be refrigerated. The flow rate of R-134a is 0.2 lb/s. The total power input is 85 Btu/s. The cooling water enters the condenser at 80°F and leaves at 115°F. The surroundings are at 210°F. Determine the overall exergy loss.

7.88 In a tetrafluoroethane (R-134a) refrigeration cycle, the superheated R-134a (state 1) enters a compressor at 253.15 K and 0.14 MPa. The R-134a (state 2) leaves the compressor at 303.15 K and 0.5 MPa, and enters a condenser, where it is cooled by cooling water. The R-134a (state 3) leaves the condenser at 299.87 K and 0.75 MPa and enters a throttling valve. The partially vaporized R-134a (state 4) leaves the valve at 0.205 MPa. The cycle is completed by passing the R-134 through an evaporator to absorb heat from the matter to be refrigerated. The R-134a (state 1) leaves the evaporator as

superheated vapor. The flow rate of R-134a is 0.16 kg/s. The total power input is 750 kW. Estimate the coefficient of performance.

7.89 A refrigerator using tetrafluoroethane (R-134a) as refrigerant operates with a capacity of 250 Btu/s. Cooling water enters the condenser at 70°F. The evaporator is at 10°F, and the condenser is at 80°F. The refrigerated space is at 20°F. Determine the ideal and actual power necessary if the compressor efficiency is 75%. Assume that the kinetic and potential energy changes are negligible.

7.90 A refrigerator using tetrafluoroethane (R-134a) as refrigerant (tetrafluoroethane) operates with a capacity of 2,500 kW. Cooling water enters the condenser at 280 K. Evaporator is at 271.92 K, and the condenser is at 299.87 K. The refrigerated space is at 280 K. Determine the ideal and actual power necessary if the compressor efficiency is 80%. Assume that the kinetic and potential energy changes are negligible.

7.91 A refrigeration cycle has a COP = 3.0. For the cycle, q_H = 2,000 kJ. Determine q_C and W_{net}, each in kJ.

7.92 A refrigeration cycle has a COP = 2.5. For the cycle, q_H = 1,500 kJ. Determine q_C and W_{net}, each in kJ.

7.93 In a pentafluoroethane (R-125) refrigeration cycle, the saturated R-125 (state 1) enters a compressor at 250 K and 3 bar. The R-125 (state 2) leaves the compressor at 320 K and 23.63 bar, and enters a condenser, where it is cooled by cooling water. The R-125 (state 3) leaves the condenser as saturated liquid at 310 K and 18.62 bar and enters a throttling valve. The partially vaporized R-125 (state 4) leaves the valve at 255 K and 3.668 bar. The cycle is completed by passing the R-125 through an evaporator to absorb heat from the matter to be refrigerated. The R-125 leaves the evaporator as saturated vapor. The evaporator temperature is 275.15 K. The flow rate of R-125 is 0.75 kg/s. The total power input is 60 kW. The cooling water enters the condenser at 293.15 K and leaves at 295.15 K. The surroundings are at 298.15 K. Determine the coefficient of performance.

7.94 A refrigerator using tetrafluoroethane (R-134a) as refrigerant operates with a capacity of 10,000 Btu/h. The refrigerated space is at 15°F. The evaporator and condenser operate with a 10°F temperature difference in their heat transfer. Cooling water enters the condenser at 70°F. Therefore, the evaporator is at 5°F, and the condenser is at 80°F. Determine the coefficient of performance if the compressor efficiency is 85%.

7.95 In a refrigeration cycle, the superheated R-134a (state 2) enters a compressor at 200°F and 90.0 psia. The R-134a (state 3) leaves the compressor at 360°F and 140 psia, and enters a condenser, where it is cooled by cooling water. The R-134a (state 4) leaves the condenser at 90.5°F and 120 psia as saturated liquid, and enters a throttling valve. The partially vaporized R-134a (state 1) leaves the valve at 100 psia. The cycle of R-134a is completed when it passes through an evaporator to absorb heat from the matter to be refrigerated. The flow rate of R-134a is 0.2 lb/s. The total power input is 85 Btu/s. The cooling

water enters the condenser at 80°F and leaves at 115°F. The surroundings are at 210°F. Determine the coefficient of performance.

7.96 In a tetrafluoroethane (R-134a) refrigeration cycle, the superheated R-134a (state 1) enters a compressor at 253.15 K and 0.14 MPa. The R-134a (state 2) leaves the compressor at 303.15 K and 0.5 MPa, and enters a condenser, where it is cooled by cooling water. The R-134a (state 3) leaves the condenser at 299.87 K and 0.75 MPa and enters a throttling valve. The partially vaporized R-134a (state 4) leaves the valve at 0.205 MPa. The cycle is completed by passing the R-134 through an evaporator to absorb heat from the matter to be refrigerated. The R-134a (state 1) leaves the evaporator as superheated vapor. The flow rate of R-134a is 0.16 kg/s. The total power input is 750 kW. Estimate the coefficient of performance.

7.97 A refrigerator using tetrafluoroethane (R-134a) as refrigerant operates with a capacity of 250 Btu/s. Cooling water enters the condenser at 70°F. The evaporator is at 10°F, and the condenser is at 80°F. The refrigerated space is at 20°F. Determine the coefficient of performance if the compressor efficiency is 75%. Assume that the kinetic and potential energy changes are negligible.

7.98 A refrigerator using tetrafluoroethane (R-134a) as refrigerant (tetrafluoroethane) operates with a capacity of 2,500 kW. Cooling water enters the condenser at 280 K. Evaporator is at 271.92 K, and the condenser is at 299.87 K. The refrigerated space is at 280 K. Determine the ideal and actual power necessary if the compressor efficiency is 80%. Assume that the kinetic and potential energy changes are negligible.

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